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PRACTICAL TREATISE ON HEAT.

THOMAS BOX.

A

PRACTICAL TREATISE

ON

HEAT,

AS APPLIED TO THE USEFUL ARTS,

FOR THE USE OF

ENGINEERS, ARCHITECTS, ETC.

BY THOMAS BOX,

AUTHOR OF 'PRACTICAL HYDRAULICS.'

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PREFACE.

THE object of the following work is to give Data, Rules, and Tables to facilitate the practical application of the Laws of Heat to the Useful Arts.

The subject has throughout been largely illustrated by Examples worked out in detail, and this has led to calculations more or less complicated; but the author's special desire to make the "principles" of the subject clear to the reader could not be so well attained by any other means.

The authorities from whom the Experimental Data, &c., are derived, are for the most part given as they occur; but Péclet's great work, 'Traité de la Chaleur,' should be more particularly mentioned.

BATH, *August*, 1868.

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PRACTICAL TREATISE ON HEAT.

CHAPTER I.

GENERAL PRINCIPLES AND FACTS.

(1.) "*Unit of Heat.*"—It is necessary to have a standard for measuring the amount of heat absorbed or evolved during any operation: in this country the standard unit is the amount of heat necessary to raise the temperature of a pound of water 1° Fahr. This amount is not quite uniform at all temperatures, as is shown by the experiments of Regnault: thus from 32° to 212° we have, by a uniform rate, $212 - 32 = 180$ units, but Regnault gives 180·9 units; and from 32° to 446, instead of $446 - 32 = 414$ units, Regnault gives 422·4 units; but for practical purposes we may in most cases assume it to be uniform.

(2.) "*Specific Heat.*"—Different bodies require very different quantities of heat to effect in them the same change of temperature. The capacity of a body for heat is termed its "Specific Heat," and may be defined as the number of units of heat necessary to raise the temperature of 1 lb. of that body 1° Fahr. Table 1 gives its value for most of the solid, liquid, and gaseous bodies commonly met with in practice. The specific heat of all bodies (except gases) appears to increase with the temperature, as we have seen to be the case with water. This is shown by Table 2, from the experiments of Dulong and Pouillet, but at ordinary temperatures the departure from uniformity is not great, and in most cases we may admit a uniform rate, having the value given by Table 1. Thus, to heat 200 lbs. of cast iron 120°, would require $200 \times 120 \times \cdot 12983 = 3116$ units of heat; the same weight of water requiring $200 \times 120 = 24000$ units, or nearly eight times the heat required for cast iron. The specific heat of air is a more intricate matter.

(3.) "*Specific Heat of Air,*" *&c.*—When air, &c., is heated in a closed vessel, the capacity or volume remaining constant, the

TABLE 1.—Of the SPECIFIC HEAT of SOLID, LIQUID, and GASEOUS BODIES, being the number of UNITS of heat necessary to heat One Pound of the Body 1° Fahrenheit.

	Specific Heat.	Authority.
Iron (wrought)	·11379	Regnault.
Ditto (cast)	·12983	,,
Copper	·09515	,,
Tin (English)	·05695	,,
Zinc	·09555	,,
Brass	·09391	,,
Lead	·03140	,,
Mercury	·03332	,,
Gold	·03244	,,
Silver	·05701	,,
Platina	·03243	,,
Bismuth	·03084	,,
Glass	·19768	,,
Marble (white)	·21585	,,
Chalk (white)	·21485	,,
Burnt Clay (white)	·18500	Gadolin.
Coal	·2777	Crawford.
Sulphur	·20259	Regnault.
Spermaceti	·3200	Irvine.
Wood (pine)	·650	Mayer.
Ditto (birch)	·480	,,
Bees-wax	·450	Gadolin.
Ice	·504	Person.
Water	1·0000	—
Olive Oil	·3096	Lavoisier.
Alcohol (s. g. ·793)	·622	Dalton.
Oil of Turpentine	·472	Despretz.
Gases under constant Pressure of 30 in. Mercury.		
Oxygen Gas	·2182	Regnault.
Hydrogen Gas	3·4046	,,
Nitrogen Gas	·2440	,,
Carbonic Acid Gas	·2164	,,
Sulphuretted Hydrogen	·2423	,,
Vapour of Water	·4750	,,
Air	·2380	,,

pressure is increased; if, on the other hand, the air be suffered to expand freely, by increase of temperature, the pressure may remain constant. Now the specific heat of air is different under these two different conditions. Considering first the specific heat under constant pressure, which is the case most generally

TABLE 2.—Of the VARIATION in the SPECIFIC HEAT of BODIES at different Temperatures.

	Between 32° and 212°.	Between 32° and 392°.	Between 32° and 572°.
Mercury (Dulong)	·0330	..	·0350
Silver ,, ...	·0557	..	·0611
Zinc ,, ...	·0927	..	·1015
Copper ,, ...	·0940	·1013	..
Iron ,, ...	·1098	·1150	·1218
Platina ,, ...	·0335	..	·0355
Antimony ,, ...	·0507	..	·0547
Glass ,, ...	·1770	..	·1900

	At 212°.	At 572°.	At 932°.	At 1292°.	At 1832°.	At 2192°.
Platina (Pouillet) ..	·03350	·03434	·03518	·03600	·03718	·03818

met with: imagine a gas-holder so well-balanced as to exert no pressure on the enclosed gas, and let it contain one pound of air at 62°, with the barometer at 29·92, or say 30 inches of mercury; its capacity (Table 32) would be 13·14 cubic feet. If heat be applied, so as to raise the temperature 1°, or to 63°, the air would be expanded slightly (14), and the gas-holder would rise to allow for it. The amount of heat required under these circumstances, according to Regnault's experiments, would be ·238 units, or the amount which would have raised the same weight of water ·238 of a degree, and this is equal to $\frac{\cdot 238}{13\cdot 14} = \cdot 01817$ units per cubic foot. If the gas under consideration were hydrogen, the capacity of the gas-holder, capable of holding one pound, would have been 189·7 cubic feet, and to effect a change of 1° in the temperature would require, according to Regnault, 3·4046 units of heat, or $\frac{3\cdot 4046}{189\cdot 7} = \cdot 018$ units per cubic foot, practically the same as a cubic foot of air. Table 1 gives the specific heat of air, gases, and vapour, under these conditions.

(4.) If the pressure were other than 30 inches (but still constant), the specific heat would be different, as is shown by the experiments of Clement and Desormes in Table 3. Here the specific heats are given for equal *volumes* instead of equal weights,

TABLE 3.—Of the SPECIFIC HEAT of AIR, from the Experiments of CLEMENT and DESORMES.

Total Pressure in inches of Mercury.	Ratio of Specific Heat at the different Pressures for equal Volumes.		Error.
	Experiment.	Calculations.	
39·6	1·215	1·152	– ·063
29·84	1·000	1·000	·000
14·92	·692	·706	+ ·013
7·44	·540	·499	– ·041
3·74	·368	·387	+ ·019

and this appears to vary directly as the square root of the pressure, as is shown by column 3 which is calculated by the rule $\sqrt{P} \times \cdot183 = R$, in which P = the pressure in inches of mercury, and R = the ratio of specific heat of air at that pressure, the specific heat at 29·84 being 1·000. Table 4 has been calculated on the same principles, thus the (col. 3) has been obtained by multiplying ·238, the specific heat at atmospheric pressure, by the square root of the density in col. 2, which again is simply proportional to the pressure in col. 1. The col. 4 is obtained by dividing col. 3 by col. 2, and col. 5 by dividing col. 4

TABLE 4.—Of the VARIATION in the SPECIFIC HEAT of AIR, at different Densities.

Total Pressure above a Vacuum in inches of Mercury.	Relative Density.	Specific Heat of equal Volumes.	Specific Heat with Pressure Constant.		Specific Heat with Volume Constant and Temperature 62°.		Cubic feet in 1 lb. of Air at 62° Fahr.
			Units per Pound at any Temperature.	Units per Cubic foot at 62°.	Units per Pound at 62°.	Units per Cubic foot at 62°.	
120	4	·476	·119	·03634	·0837	·02556	3·275
60	2	·336	·168	·02690	·1184	·01808	6·55
30	1	·238	·238	·01817	·1674	·01280	13·1
15	$\frac{1}{2}$	·168	·336	·01284	·2367	·00903	26·2
7·5	$\frac{1}{4}$	·119	·476	·00908	·3348	·00639	52·4
3·75	$\frac{1}{8}$	·084	·672	·00642	·4734	·00452	104·8
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)

by col. 8. The general formula for the specific heat of gases with constant pressure, is $\frac{S \times \sqrt{30}}{\sqrt{P}} = S'$, in which S = the specific heat at 30 inches of mercury, as in Table 1 and in col. 1 of Table 5, P = the total pressure in inches of mercury, and S' = the specific heat under that pressure. For air this becomes $\frac{1 \cdot 303}{\sqrt{P}} = S'$, and col. 2 in Table 6 has been thus calculated.

TABLE 5.—Of the SPECIFIC HEAT of GASES and VAPOUR.

	Specific Heat of 1 lb. of Gas under a Constant Pressure of 29·92 inches of Mercury (Regnault).	Ratio of the Specific Heat under Constant Pressure, to the Specific Heat with Constant Volume (Dulong).	Specific Heat of 1 lb. of Gas taken with the Barometer at 29·92 inches, the Volume remaining Constant.
Oxygen Gas	·2182	1·415	·1542
Hydrogen Gas	3·4046	1·407	2·4200
Nitrogen Gas	·2440	..	·1717
Carbonic Acid Gas	·2164	1·338	·1617
Carbonic Oxyde Gas	·2479	1·427	·1737
Sulphuretted Hydrogen ..	·2423	..	·1825
Vapour of Water	·4750	..	·3624
Atmospheric Air	·2380	1·421	·1674
	(1)	(2)	(3)

(5.) When air and gases are heated in closed vessels, expansion being prevented, the pressure is increased, and the specific heat is less than when the pressure is constant and expansion permitted. The experiments of Dulong give the ratio of specific heat under these two different conditions for several of the most important gases, as per col. 2 in Table 5; thus for air the ratio is 1·421 to 1, and we have $\frac{\cdot 238}{1 \cdot 421} = \cdot 1674$ for the specific heat of air in a closed vessel, and under a pressure of 30 inches of mercury. The general rule for the specific heat of gases with constant volume is $\frac{S}{R} \times \frac{\sqrt{30}}{\sqrt{P}} = S''$, in which R is the ratio, as per col. 2 in Table 5, S'' = the specific heat with constant volume, and S and P the same as before. For air the rule becomes $\frac{\cdot 917}{\sqrt{P}} = S''$: thus with a pressure of 120 inches of mercury, the specific heat of air in a closed vessel would be $\frac{\cdot 917}{\sqrt{120}}$, or $\frac{\cdot 917}{10\cdot 95} = \cdot 0837$, as per col. 6 in Table 4, and col. 3 in Table 6. We may deduce the specific heat under constant volume from that under constant pressure by the rule $S'' = S' - \frac{\cdot 070216}{G}$, in which G = the specific gravity of the gas, air being 1·0, and S′ and S″ the same value as before and thus in Table 5 we have obtained the numbers, in col. (3), for gases, &c., whose ratio is not given by the experiments of Dulong. This rule is given by Bunsen.

TABLE 6.—Of the SPECIFIC HEAT of AIR at different Pressures.

Total Pressure in inches of Mercury.	Specific Heat with		Total Pressure in inches of Mercury.	Specific Heat with	
	Pressure Constant, and Volume Variable with Change of Temperature.	Volume Constant, and Pressure Variable with Change of Temperature.		Pressure Constant, and Volume Variable with Change of Temperature.	Volume Constant, and Pressure Variable with Change of Temperature.
1	1·303	·9170	70	·1558	·1096
5	·5827	·4101	75	·1504	·1059
10	·4121	·2900	80	·1457	·1025
15	·3360	·2367	85	·1413	·0995
20	·2914	·2050	90	·1374	·0967
25	·2606	·1834	100	·1306	·0919
			110	·1242	·0874
30	·2380	·1674	120	·1190	·0837
			130	·1143	·0804
35	·2203	·1550	140	·1101	·0775
40	·2061	·1450	150	·1064	·0749
45	·1943	·1367	180	·0971	·0684
50	·1843	·1297	210	·0899	·0633
55	·1757	·1236	240	·0841	·0592
60	·1680	·1184	270	·0793	·0558
65	·1616	·1138	300	·0752	·0529

(6). "*Latent Heat of Liquefaction.*"—When a body passes from the solid to the liquid state, it absorbs a large amount of heat without changing its own temperature, the heat thus absorbed becoming latent, and insensible to the thermometer. This is termed the "latent heat of liquefaction," and may be defined as the number of units of heat absorbed by one pound of the solid, in passing to the liquid state. When the process is reversed, the liquid congealing, freezing, or passing back again to the solid state, the same amount of heat is emitted or restored. Thus when ice is heated to 32°, it begins to melt, the temperature remaining fixed until the last particle of ice is melted; and during this process 142·4 units are absorbed, being the amount that would have raised the same weight of water 142°·4; but the ice itself, having, by Table 1, a specific heat of ·504, would have had its temperature raised $\frac{142\cdot4}{\cdot504} = 281^\circ$ if the heat had not become latent. The "latent heat," however, is not 281, but 142·4 *units*. Table 7 gives the latent heat of liquefaction,

TABLE 7.—Of the LATENT HEAT of LIQUEFACTION, being the number of Units of Heat absorbed by One Pound of different Bodies in changing their state from Solid to Liquid.

	Latent Heat in Units.	Increase of Temperature in the Body if Heat had not been absorbed in melting.	Authority.
		°	
Ice to Water ..	142·4	281	Person.
Sulphur	16·8	83	,,
Tin	25·6	450	,,
Lead	9·7	309	,,
Zinc	50·6	530	,,
Bismuth	22·8	740	,,
Silver	38·0	665	,,
Cast Iron	233·0	1574	Clement.
Beeswax	78·7	175	Irvine.
Spermaceti ..	46·4	145	,,

the col. 3 being obtained by dividing the latent heat by the specific heat.

(7) "*Latent Heat of Vapourization.*"—When a liquid changes its state to the gaseous or vapourous state, another large amount of heat becomes latent, and this is termed the "latent heat of vapourization;" and it may be defined as the number of units of heat absorbed by one pound of a liquid in the act of passing to the gaseous state. Thus a pound of water at 212°, passing to steam at 212°, absorbs, according to Regnault, as much heat as would have raised the temperature of the water 966° if it had not become latent. Again, a pound of alcohol absorbs

457 units in the act of vapourization, or the amount of heat that would have raised a pound of water 457; but alcohol having by Table 1 a specific heat of ·622, its own temperature would have been raised $\frac{457}{\cdot 622} = 735^\circ$ if heat had not become latent. Table 8 gives the latent heat of vapourization for a few of the liquids most commonly met with in practice.

TABLE 8.—Of the LATENT HEAT of VAPOURIZATION, being the number of Units of Heat required to convert Liquids from their respective Boiling-points to Vapour, under a pressure of 30 in. Mercury.

	Latent Heat in Units.	Increase of Temperature of Liquid if Heat had not become Latent.	
		°	
Water	966	966	Regnault.
Alcohol	457	735	Ure.
Ether	313	473	,,
Oil of Turpentine	184	390	,,
Naphtha	184	—	,,

(8) Table 9 shows that the temperature of the boiling-point of water, varies with the pressure; and it has been found by experiment, that the amount of heat which becomes latent during vapourization, varies with the temperature at which it is effected,

TABLE 9.—Of the BOILING POINTS of LIQUIDS, at Atmospheric Pressure.

		Temp. Fahr.	
		°	
Ether, Sulphuric, sp. gr.	·7365 ..	100	G. Lussac.
Alcohol ,,	·813 ..	173	Ure.
Muriatic Acid ,,	1·047 ..	222	Dalton.
Nitric Acid ,,	1·16 ..	220	,,
Sulphuric Acid ,,	1·3 ..	240	,,
,, ,, ,,	1·85 ..	620	,,
Oil of Turpentine	..	316	Ure.
Naphtha	..	306	,,
Sulphur	..	570	,,
Linseed Oil..	..	600	—
Mercury	..	652	Mean of 6.
Water	..	212	—

but that the *total* amount of heat necessary to raise the liquid from a low temperature and then evaporate it, is constant. Thus the heat required to raise a pound of water from 0° to 212° and then evaporate it, is 212 + 966 = 1178 units. Table 10 shows that if the atmospheric pressure were reduced to 15 inches of mercury in the barometer, water would boil at 180° instead of 212°, but the units of heat from 0° would still be 1178; and, as 180° only were required to raise the water to its new boiling-point, the latent heat of vapourization must be 1178 − 180 = 998 units, instead of 966. Again, at a pressure of 60 lbs. per square inch above the atmosphere, Table 11 shows that water must be heated to 307° before ebullition commences; but the latent heat of vapourization will be proportionately diminished, and will in that case become 1178 − 307 = 871 units, instead of 966 units.

TABLE 10.—Of the TEMPERATURE of the BOILING-POINT of VARIOUS LIQUIDS under different Pressures, deduced from the Experiments of REGNAULT.

Liquid	Pressure in inches of Mercury.			
	30	15	10	5
	Temperature of Ebullition.			
	°	°	°	°
Water	212	180	162	134
Alcohol	173	146	127	101
Ether	100	62	55	17
Oil of Turpentine ..	316	271	248	210

See also Tables 12, 22, &c.

For convenience of calculation in other parts of this work, it is assumed in the above that water could be reduced to 0° without passing to the solid state, or to ice, where, as we have seen, its specific heat is altered. This, of course, is fictitious: the real amount of heat required to convert a pound of ice at 0° to steam at 212° is 1304·5 units, as follows:—

A pound of ice at 0° to ice at 32°, = 32 × ·504	=	16·1	units
„ ice at 32° to water at 32° ..	=	142·4	„
„ water at 82° to water at 212°	=	180·0	„
„ „ 212° to steam at 212°	=	966·0	„
Total	=	1304·5	„

TABLE 11.—Of the TEMPERATURE and VOLUME of STEAM at different Pressures, calculated from the Experiments of REGNAULT, &c.

Pressure above the Atmosphere in lbs. per square in.	Temp. Fahr.	Cubic feet of Steam from one Cubic foot of Water.	Pressure above the Atmosphere in lbs. per square in.	Temp. Fahr.	Cubic feet of Steam from one Cubic foot of Water.	Pressure above the Atmosphere in lbs. per square in.	Temp. Fahr.	Cubic feet of Steam from one Cubic foot of Water.
0	212°	1640	44	291°	446	88	330°	264
1	215	1544	45	292	439	89	,,	261
2	219	1456	46	294	432	90	331	258
3	222	1378	47	295	426	91	332	256
4	224	1310	48	296	419	92	333	254
5	227	1247	49	297	413	93	,,	252
6	230	1190	50	298	407	94	334	250
7	232	1138	51	299	401	95	,,	248
8	235	1089	52	300	394	96	335	246
9	237	1048	53	301	387	97	336	244
10	239	1008	54	302	384	98	,,	242
11	241	969	55	303	379	99	337	240
12	244	936	56	304	374	100	338	238
13	246	904	57	305	369	102	339	234
14	248	875	58	,,	364	104	340	230
15	250	846	59	306	360	105	341	228
16	252	820	60	307	355	106	342	226
17	253	796	61	308	350	108	343	223
18	255	774	62	309	346	110	344	220
19	257	743	63	310	342	115	347	212
20	259	732	64	311	338	120	350	203
21	260	713	65	312	334	125	353	197
22	262	694	66	313	330	130	355	190
23	264	676	67	314	326	135	358	184
24	265	660	68	,,	322	140	361	179
25	267	644	69	315	319	145	363	174
26	268	630	70	316	315	150	366	169
27	270	615	71	317	312	160	370	159
28	271	602	72	318	309	170	375	151
29	272	589	73	,,	306	180	380	144
30	274	576	74	319	302	190	384	138
31	275	564	75	320	299	200	388	132
32	277	553	76	321	296	210	392	126
33	278	542	77	322	293	220	396	121
34	279	532	78	,,	290	230	399	116
35	281	521	79	323	287	240	403	112
36	282	512	80	324	284	250	406	108
37	283	503	81	325	282	260	409	104
38	284	494	82	,,	279	270	413	101
39	285	485	83	326	276	280	416	98
40	286	476	84	327	273	290	419	95
41	288	468	85	328	271	300	422	91
42	289	460	86	,,	268			
43	290	453	87	329	266			

Table 12.—Of the Elastic Force of Vapour of Water in inches of Mercury, calculated from the Experiments of Regnault.

Temp. Fahr.	Force in inches.	Temp. Fahr.	Force in inches.	Temp. Fahr.	Force in inches.	Temp. Fahr.	Force in inches.	Temp. Fahr.	Force in inches.
°		°		°		°		°	
0	·044	46	·311	92	1·501	138	5·565	184	16·68
1	·046	47	·323	93	1·548	139	5·710	185	17·05
2	·048	48	·335	94	1·596	140	5·858	186	17·42
3	·050	49	·348	95	1·646	141	6·010	187	17·81
4	·052	50	·361	96	1·697	142	6·165	188	18·20
5	·054	51	·374	97	1·751	143	6·324	189	18·60
6	·057	52	·388	98	1·806	144	6·488	190	19·01
7	·060	53	·403	99	1·862	145	6·655	191	19·42
8	·062	54	·418	100	1·918	146	6·825	192	19·83
9	·065	55	·433	101	1·976	147	7·000	193	20·26
10	·068	56	·449	102	2·036	148	7·177	194	20·68
11	·071	57	·465	103	2·098	149	7·360	195	21·12
12	·074	58	·482	104	2·162	150	7·546	196	21·57
13	·078	59	·500	105	2·227	151	7·736	197	22·03
14	·082	60	·518	106	2·293	152	7·930	198	22·50
15	·086	61	.537	107	2·361	153	8·128	199	22·97
16	·090	62	·556	108	2·431	154	8·330	200	23·46
17	·094	63	·576	109	2·503	155	8·536	201	23·95
18	·098	64	·596	110	2·577	156	8·746	202	24·45
19	·103	65	·617	111	2·653	157	8·960	203	24·95
20	·108	66	·639	112	2·731	158	9·177	204	25·47
21	·113	67	·661	113	2·811	159	9·400	205	26·00
22	·118	68	·685	114	2·893	160	9·628	206	26·53
23	·123	69	·708	115	2·977	161	9·861	207	27·08
24	·129	70	·733	116	3·063	162	10·099	208	27·64
25	·135	71	·759	117	3·151	163	10·342	209	28·19
26	·141	72	·785	118	3·241	164	10·590	210	28·76
27	·147	73	·812	119	3·333	165	10·843	211	29·34
28	·153	74	·840	120	3·427	166	11·101	212	29·92
29	·160	75	·868	121	3·523	167	11·36	213	30·52
30	·167	76	·897	122	3·621	168	11·63	214	31·13
31	·174	77	·927	123	3·721	169	11·90	215	31·75
32	·181	78	·958	124	3·824	170	12·18	216	32·38
33	·188	79	·990	125	3·930	171	12·46	217	33·02
34	·196	80	1·023	126	4·039	172	12·75	218	33·67
35	·204	81	1·057	127	4·151	173	13·05	219	34·33
36	·212	82	1·092	128	4·265	174	13·35	220	35·01
37	·220	83	1·128	129	4·382	175	13·66	221	35·68
38	·229	84	1·165	130	4·502	176	13·96	222	36·37
39	·238	85	1·203	131	4·625	177	14·28	223	37·08
40	·247	86	1·242	132	4·752	178	14·60	224	37·80
41	·257	87	1·282	133	4·881	179	14·93	225	38·53
42	·267	88	1·323	134	5·012	180	15·27	226	39·27
43	·277	89	1·366	135	5·146	181	15·61	227	40·02
44	·288	90	1·401	136	5·283	182	15·96	228	40·79
45	·299	91	1·455	137	5·423	183	16·32	229	41·56

TABLE 12—*continued.*

Temp. Fahr. °	Force in inches.	Temp. Fahr. °	Force in inches.	Temp. Fahr. °	Force in inches.	Temp. Fahr. °	Force in inches.	Temp. Fahr. °	Force in inches.
230	42·34	274	91·18	318	177·9	362	320·5	406	539·5
231	43·14	275	92·67	319	180·5	363	324·6	407	545·5
232	43·95	276	94·18	320	183·1	364	328·7	408	551·6
233	44·78	277	95·72	321	185·7	365	332·8	409	557·7
234	45·62	278	97·27	322	188·3	366	336·9	410	563·9
235	46·47	279	98·84	323	191·0	367	341·2	411	570·1
236	47·32	280	100·4	324	193·7	368	345·4	412	576·4
237	48·20	281	102·0	325	196·4	369	349·7	413	582·8
238	49·08	282	103·6	326	199·2	370	354·0	414	589·2
239	49·98	283	105·3	327	201·9	371	358·4	415	595·7
240	50·89	284	107·00	328	204·8	372	362·8	416	602·2
241	51·83	285	108·7	329	207·6	373	367·2	417	608·8
242	52·77	286	110·4	330	210·5	374	371·8	418	615·4
243	53·72	287	112·1	331	213·4	375	376·3	419	622·1
244	54·69	288	113·9	332	216·4	376	380·9	420	628·8
245	55·68	289	115·7	333	219·4	377	385·5	421	635·6
246	56·68	290	117·5	334	222·4	378	390·2	422	642·5
247	57·69	291	119·3	335	225·4	379	394·9	423	649·4
248	58·71	292	121·2	336	228·5	380	399·6	424	656·3
249	59·75	293	123·05	337	231·6	381	404·4	425	663·3
250	60·81	294	124·9	338	234·7	382	409·3	426	670·4
251	61·89	295	126·8	339	237·8	383	414·1	427	677·5
252	62·98	296	128·8	340	241·1	384	419·0	428	684·7
253	64·09	297	130·7	341	244·3	385	424·0	429	691·9
254	65·21	298	132·7	342	247·6	386	429·0	430	699·2
255	66·35	299	134·7	343	250·9	387	434·1	431	706·5
256	67·50	300	136·8	344	254·3	388	439·2	432	713·9
257	68·66	301	138·9	345	257·6	389	444·4	433	721·4
258	69·85	302	141·0	346	261·1	390	449·6	434	728·9
259	71·05	303	143·1	347	264·5	391	454·9	435	736·5
260	72·27	304	145·2	348	268·0	392	460·2	436	744·1
261	73·50	305	147·4	349	271·5	393	465·5	437	751·8
262	74·76	306	149·6	350	275·0	394	470·9	438	759·6
263	76·03	307	151·8	351	278·6	395	476·4	439	767·4
264	77·31	308	154·0	352	282·3	396	481·9	440	775·3
265	78·62	309	156·3	353	285·9	397	487·4	441	783·2
266	79·93	310	158·6	354	289·6	398	493·0	442	791·2
267	81·27	311	160·9	355	293·4	399	498·7	443	799·3
268	82·63	312	163·2	356	297·1	400	504·4	444	807·4
269	84·01	313	165·6	357	300·9	401	510·1	445	815·6
270	85·41	314	168·0	358	304·7	402	515·9	446	823·9
271	86·83	315	170·5	359	308·6	403	521·7	447	832·2
272	88·26	316	172·9	360	312·6	404	527·6	448	840·6
273	89·72	317	175·4	361	316·5	405	533·5	449	849·0

But this difference will not affect the correctness of our calculation; for instance, the amount of heat to convert water at 32 to steam is by the above, $180 + 966 = 1146$ units, and by the other method, $1178 - 32 = 1146$ units also.

(9.) These results, however, will be slightly modified by the fact that the latent heat of water itself varies with the temperature (1), and the total heat from 32° will be more accurately given, according to the experiments of Regnault, by the formula, $H = 1081{\cdot}4 + {\cdot}305\ t$, in which H = the total heat from 32°, at the temperature t; thus at 212° we have $1081{\cdot}4 + ({\cdot}305 \times 212) = 1146{\cdot}06$ units.

The heat that becomes latent will be given precisely by the formula $\angle = 1115{\cdot}2 - {\cdot}708\ t$; $\angle$ being the latent heat at the temperature t.

(10.) "*Boiling-point of Liquids.*"—When the temperature of any liquid is raised to a certain point, the liquid passes off in a state of vapour, and the temperature remains constant. This temperature varies however, with the pressure of the atmosphere or vapour on the surface of the liquid. Table 9 gives the boiling-points of liquids at ordinary atmospheric pressures, and Table 10 the variation in temperature with reduced pressure. Tables 11 and 12 give the temperatures with increased pressure.

The boiling-point is not affected by foreign bodies in the water, &c., so long as the body does not combine chemically with the water; but a great number of salts, &c., do so combine, and the boiling-point is raised, as shown by Table 13, from the experiments of M. Legrand and others. It has been found that the *vapour* produced at the surface of a saline solution is that of pure water, and has the same temperature of 212° under 30 inches of mercury, although the temperature of the solution itself may be much higher.

(11.) "*Expansion.*"—All bodies, whether solid, liquid, or gaseous, are expanded or increased in volume by heat, but in very different proportions, as shown by Table 14. In the case of solids the expansion may be estimated by the increase in length, or by the increase in volume. Let us imagine a very expansible solid, such that by a given change of temperature its length would be doubled, say from one foot to two feet; now if the body were a cube, expansion taking place equally in all directions, it is obvious that, while before expansion it measured one cubic foot, it would, after expansion, measure $2 \times 2 \times 2 = 8$ cubic feet. Estimating by length, therefore, we should say that

TABLE 13.—Of the BOILING-POINTS of SOLUTIONS of VARIOUS SALTS, at ordinary Atmospheric Pressure, from the Experiments of M. LEGRAND and others.

	Temperature of Ebullition.	Weight of Salt in 100 lbs. of Water.
	°	lbs.
Saturated solution of Chlorate of Potassa	219·56	61·50
,, ,, Carbonate of Soda	220·28	48·50
,, ,, Phosphate of Soda	221·9	113·30
,, ,, Chloride of Potassium	226·94	59·40
,, ,, ,, Soda	227·12	41·20
,, ,, Neutral Tartrate of Potassa ..	238·40	296·20
,, ,, Nitre, or Saltpetre	240·62	335·10
,, ,, Nitrate of Soda	249·80	224·80
,, ,, Acetate of Soda	255·90	209·00
,, ,, Carbonate of Potash	275·00	205·00
,, ,, Nitrate of Lime	303·80	362·20
,, ,, Acetate of Potassa	336·20	798·20
,, ,, Chloride of Calcium	355·10	325·00
,, ,, Nitrate of Ammonia	356·00	Infinite
,, ,, Common Salt	226·0	36·37
Not saturated solution of Common Salt	224·9	33·34
,, ,, ,, ...	223·7	30·30
,, ,, ,, ...	222·5	27·28
,, ,, ,, ...	221·4	24·25
,, ,, ,, ...	220·2	21·22
,, ,, ,, ...	219·0	18·18
,, ,, ,, ...	217·9	15·15
,, ,, ,, ...	216·7	12·12
,, ,, ,, ...	215·5	9·09
,, ,, ,, ...	214·4	6·06
,, ,, ,, ...	213·2	3·03
,, ,, ,, ...	212·0	0·00

the comparative lengths at the two temperatures were as 1 to 2, but by volume as 1 to 8, and in all cases the increase in volume is the cube of the increase in length.

For the very small dilatation of solids, such as are met with by experience, the expansion in volume may without sensible error be taken at three times the linear dilatation, for a cube has three dimensions, length, breadth, and height, and if each of these dimensions be increased by a very small amount, it is evident that the expansion of the cube in volume is very nearly three times the linear expansion.

"*Contraction of Metals in Casting.*"—The contraction which

TABLE 14.—Of the EXPANSION of BODIES by HEAT for 1° Fahrenheit, being the expansion per degree between 32° and 212°, the Volume at 32° being 1·0.

	Expansion for 1° Fahrenheit.		No. of Authorities.
	In Length.	In Volume.	
Fire Brick	·000002349	·000007047	1
Marble (black)	·000002407	·000007420	1
White Deal	·000002556	·000007669	2
Brick, stock	·000003057	·000009170	1
Marble (Carrara)	·000003633	·000010900	1
Granite (Aberdeen grey)	·000004386	·000013157	1
Glass tube	·000004567	·000013701	5
Platina	·000004835	·000014506	2
Slate (Penrhyn)	·000005764	·000017290	1
Cast Iron	·000006167	·000018501	3
Steel, rod	·000006441	·000019324	4
Wrought Iron	·000006689	·000020067	4
Iron Wire	·000007430	·000022290	2
Roman Cement	·000007972	·000023915	1
Copper	·000010088	·000030264	4
Brass, cast	·000010417	·000031250	1
,, plate	·000010450	·000031350	3
,, wire	·000010723	·000032170	1
Silver	·000011121	·000033364	6
Tin	·000013102	·000039307	4
Lead	·000015876	·000047628	2
Zinc, hammered	·000017268	·000051806	1
Mercury	..	·000100540	7
,, (in glass)	..	·000086839	1
*Water (40° to 212°)	..	·0002517	1
,, ,, (in glass)	..	·0002380	1
Alcohol (30° to 100°) ,,	..	·0006318	1
Linseed Oil (32° to 212°) (in glass)	..	·0004030	1

metals experience in cooling down from their melting points to ordinary temperatures is very considerable, and allowance has to be made for it in fixing the size of the pattern.

Table 15 gives the result of practical observations on this subject; thus a cast-iron girder, 20 feet long, must have a pattern ·1246 × 20 = 2·492 inches longer than itself; but a pattern 20 feet long would give a casting ·1236 × 20 = 2·472 inches shorter than itself. For practical purposes ⅛th of an inch to a foot is a good approximation.

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* See Table 18.

TABLE 15.—Of the CONTRACTION of METALS in CASTING.

	Length of Pattern.		Contraction.			
			Total in inches.	Per foot.		
				Of Pattern.	Of Casting.	
	ft.	in.		inches.	inches.	
Cast-iron girder	21	$8\frac{3}{4}$	$2\frac{11}{16}$	·1236	·1246	
,, ,, ...	16	9	2·05	·1225	·1236	
Gun-metal bar	5	$4\frac{5}{8}$	1·0	·18568	·1886	Maximum.
,, ...	5	$7\frac{15}{16}$	·936	·1653	·1676	
,, ...	,,		·97	·1713	·1737	
,, ...	6	$0\frac{1}{4}$	1·0	·1661	·1684	
,, ...	5	$6\frac{1}{16}$	·92	·1671	·1695	
,, ...	,,		·90	·1635	·1657	
,, ...	,,		·88	·1598	·1620	
,, ...	,,		·84	·1526	·1545	Minimum.
,, ...	..		..	·1607	·1632	Mean of 8.
Copper and Tin, Copper 113, Tin 10	5	$6\frac{3}{16}$	·895	·1623	·1645	Maximum.
,, ,,	,,		·880	·1595	·1617	
,, ,,	,,		·880	·1595	·1617	
,, ,,	,,		·855	·1550	·1570	Minimum.
,, ,,	,,		..	·1591	·1612	Mean of 4.
Yellow Brass	2	$9\frac{1}{8}$	·5	·1811	·1839	
Copper	7	$10\frac{7}{8}$	1·54	·1948	·1980	Minimum.
,, ...	7	$5\frac{1}{8}$	1·465	·1972	·2005	
,, ...	,,		1·465	·1972	·2005	Maximum.
,, ...	..		..	·1964	·1996	Mean of 4.
Lead	2	0	·21	·1050	·1059	
Zinc, cast in iron mould ..	2	$0\frac{3}{16}$	·455	·2257	·2301	Minimum.
,, ,, ..	,,		·465	·2307	·2352	Maximum.
,, ,, ..	..		..	·2282	·2326	Mean of 2.

The contraction of wheels is anomalous, as is shown by Table 16. The great irregularities in the *apparent* contraction arise in great part from the practice of "rapping" the pattern in the sand, to make it an easy fit, and enable it to be drawn out with facility. This is most influential in small heavy wheels, of great width of face: in some cases, and in rough hands, the casting of a small and heavy pinion may be quite the full size of the pattern. The allowance to be made is, therefore, not uniform, but must be fixed by judgment. But besides this, a wheel, &c., is not so free to contract as a straight bar, and in any case its contraction will be rather less.

TABLE 16.—Of the CONTRACTION in CASTING SPUR WHEELS in CAST IRON.

Extreme diameter of Wheel Casting.	Pitch in inches.	Width of Teeth in inches.	Contraction.		
			Total in inches.	Per foot.	
				Of Casting.	Of Pattern.
ft. in.				inches.	inches.
10 2⅜	3½	12	1·08	·1059	·1040
6 2$\frac{9}{16}$	3¼	9	·54	·0893	·0886
6 1⅜	3½	11	·375	·0613	·0610
5 5$\frac{9}{16}$	3½	11	·345	·0631	·0628
2 11⅞	3½	12	·11	·03896	·03884
2 4$\frac{15}{16}$	3¼	9	·115	·0397	·0396

(12.) "*The Expansion of Liquids.*"—The expansion of liquids must be estimated by the increase in *volume*. Referring to the former illustration (11), we may suppose that a cubic foot of the liquid is contained in a vessel that *does not itself expand with heat*, but of such a *height* as to allow the liquid to expand in that direction only. When, by expansion, the cubic foot of liquid becomes 8 cubic feet, it is obvious that the vessel, whose length and breadth is fixed, must be 8 feet high to hold the expanded liquid, and thus the *linear* dilatation is in fact the actual expansion in *volume*.

But if the vessel is itself expansible, the observed expansions are apparent only, not real and absolute, being in fact the

TABLE 17.—Of the VARIATION in the EXPANSIONS of BODIES at different Temperatures, from the Experiments of DULONG.

	32° to 212°.	32° to 392°.	32° to 572°.
	Linear Expansion for 1° Fahrenheit.		
Iron	·000006567	..	·000008158
Copper ..	·000009545	..	·000010462
Glass ..	·000004785	·000005125	·000005616
	Expansion in Volume for 1° Fahrenheit.		
Mercury ..	·0001001	·00010241	·00010482

7c

difference between the expansion of the liquid and that of the vessel containing it. Thus from Table 14 the expansion of glass in volume is ·000013701, and the *absolute* expansion of mercury is ·00010054, the *apparent* expansion of mercury in a glass vessel (such as a thermometer bulb, &c.) will therefore be ·00010054 – ·000013701 = ·000086839, as per Table 14.

The expansion of water is exceptional and anomalous. It attains a minimum volume and a maximum density at 40°, and a departure from that temperature, in either direction, is accompanied by expansion, so that 8° or 10° of cold produces

TABLE 18.—Of the VOLUME, SPECIFIC GRAVITY, EXPANSION, and WEIGHT of WATER at different Temperatures.

Temp. Fahr.	Volume.	Specific Gravity.	Weight of a Cubic foot in pounds.	Expansion for 1° between the different Temperatures.
20°	1·0012000	·99880	62·33	·0000822
30	1·0003780	·99962	62·38	·0000378
40	1·0000000	1·00000	62·408	·0000129
42	1·0000258	·99997	62·406	·0000486
52	1·0005123	·99950	62·377	·0000895
62	1·0014070	·9986	62·321	·0001216
72	1·002627	·9974	62·25	·0001516
82	1·004143	·9959	62·15	·0001758
92	1·005901	·9941	62·04	·0002010
102	1·007911	·9921	61·92	·0002239
112	1·010150	·9900	61·78	·000246
122	1·01261	·9875	61·63	·000266
132	1·01527	·9850	61·47	·000287
142	1·01814	·9822	61·30	·000306
152	1·02120	·9792	61·11	·000323
162	1·02443	·9761	60·92	·000345
172	1·02788	·9729	60·72	·000360
182	1·03148	·9695	60·5	·000378
192	1·03526	·9659	60·28	·000396
202	1·03922	·9622	60·05	·000411
212	1·04333	·9585	59·82	·000434
230	1·05115	·9513	59·37	·000464
250	1·06043	·9430	58·85	·000498
275	1·07289	·9321	58·17	·000562
300	1·08693	·9200	57·42	·000573
350	1·11560	·8963	55·94	·000656
400	1·14840	·8708	54·34	·000718
450	1·18430	·8444	52·70	·000780
500	1·22330	·8175	51·02	·000866
600	1·30990	·7634	47·64	

about the same amount of expansion as 8° or 10° of heat. This is shown by Table 18, which is calculated by Tredgold's rule—

$$\frac{5}{3} \log. (t - 40) + \bar{6}\cdot 910909 = \text{the log. of the Expansion.}$$

(13.) The expansions of solids and liquids are not equal for equal increments of heat, but increase with the temperature, as is shown by Table 17, from the experiments of Dulong; but for practical purposes we may in most cases admit a uniform rate, having the value given by Table 14.

(14.) "*Expansion of Gases.*"—It has been found by experiment that all the gases, dry air, and even vapours *out of contact with their generating fluids* expand alike, with the same change of temperature, or very nearly so. Regnault's experiments give the expansion from 32° to 212° at ·00367 of the bulk at 32°. The general formula becomes $V' = V \times \frac{458\cdot 4 + t'}{458\cdot 4 + t}$;

in which V = volume of gas, &c., at the temperature t,

and V' = ,, at the new temperature t'.

Thus air whose volume at 32° = 1·0000 will have at 2500° a volume of $1\cdot 0000 \times \frac{458\cdot 4 + 2500}{458\cdot 4 + 32} = 6\cdot 032$. Table 19 has been caculated by this rule, and Table 20 gives a comparison of its results with the experiments of Dulong and Petit.

(15.) When the pressure is not constant, the volume may be calculated by the law of Marriotte, namely, that *the volume of any gas varies in the inverse ratio of the pressure, the temperature remaining constant.* The pressure here meant is the total pressure above a vacuum. Thus a cubic foot of air, in ordinary cases, has the pressure of the atmosphere upon it to begin with, say 15 lbs. (nearly) per square inch, and its volume, under a pressure of say 45 lbs. per square inch above atmosphere, will be $1 \times \frac{15}{15 + 45} = \cdot 25$ cubic foot. Practical men have generally to deal with pressures above the atmosphere, and are apt to forget to add the pressure of the atmosphere itself, and thereby make serious errors in calculation.

(16.) When there is a change both in temperature and pressure, the rule becomes $V' = V \times \frac{P}{P'} \times \frac{458\cdot 4 + t'}{458\cdot 4 + t}$, in which V, P, and t are the Volume, Pressure, and temperature in one case, and V', P', and t' the Volume, Pressure, and temperature in another case. Thus 10 cubic feet of air at ordinary atmospheric

TABLE 19.—Of the VOLUME and WEIGHT of DRY AIR at DIFFERENT TEMPERATURES under a constant Atmospheric Pressure of 29·92 inches of Mercury in the Barometer, the Volume at 32° being 1.

Temp.	Volume.	Weight of a Cubic foot in pounds.	Temp.	Volume.	Weight of a Cubic foot in pounds.
0°	·935	·0864	550°	2·056	·0384
12	·960	·0842	600	2·158	·0376
22	·980	·0824	650	2·260	·0357
32	1·000	·0807	700	2·362	·0338
42	1·020	·0791	750	2·464	·0328
52	1·041	·0776	800	2·566	·0315
62	1·061	·0761	850	2·668	·0303
72	1·082	·0747	900	2·770	·0292
82	1·102	·0733	950	2·872	·0281
92	1·122	·0720	1000	2·974	·0268
102	1·143	·0707	1100	3·177	·0254
112	1·163	·0694	1200	3·381	·0239
122	1·184	·0682	1300	3·585	·0225
132	1·204	·0671	1400	3·789	·0213
142	1·224	·0660	1500	3·993	·0202
152	1·245	·0649	1600	4·197	·0192
162	1·265	·0638	1700	4·401	·0183
172	1·285	·0628	1800	4·605	·0175
182	1·306	·0618	1900	4·809	·0168
192	1·326	·0609	2000	5·012	·0161
202	1·347	·0600	2100	5·216	·0155
212	1·367	·0591	2200	5·420	·0149
230	1·404	·0575	2300	5·624	·0142
250	1·444	·0559	2400	5·828	·0138
275	1·495	·0540	2500	6·032	·0133
300	1·546	·0522	2600	6·236	·0130
325	1·597	·0506	2700	6·440	·0125
350	1·648	·0490	2800	6·644	·0121
375	1·689	·0477	2900	6·847	·0118
400	1·750	·0461	3000	7·051	·0114
450	1·852	·0436	3100	7·255	·0111
500	1·954	·0413	3200	7·459	·0108

TABLE 20.—Of the EXPANSION of DRY AIR by HEAT.

Temperature.	Volume by Experiment. Dulong and Petit.	Volume by Calculation.	Temperature.	Volume by Experiment. Dulong and Petit.	Volume by Calculation.
− 32·8°	·8650	·8678	392°	1·7389	1·734
+ 32	1·0000	1·0000	482	1·9198	1·918
212	1·3750	1·367	572	2·0976	2·101
302	1·5576	1·551	680	2·3125	2·322

pressure, say 15 lbs. per square inch, and temperature 60°, would, if heated to 200°, and under a pressure of 40 lbs. per square inch above the atmosphere, or $15 + 40 = 55$ lbs. above a vacuum, become $10 \times \frac{15}{55} \times \frac{458 \cdot 4 + 200}{458 \cdot 4 + 60} = 3 \cdot 7$ cubic feet.

(17.) "*Expansion of Moist Air.*"—When the vapour of water or other liquid, is present in the air or gas, another element becomes necessary in the calculation of its bulk at different temperatures, namely the elastic force of the vapour at the given temperature, which is given for water by Table 12. The rule then becomes $V' = V \times \frac{P' + F'}{P + F} \times \frac{458 \cdot 4 + t'}{458 \cdot 4 + t}$, in which V, P, F, and t represent the Volume, Pressure, elastic Force of vapour, and temperature in one case, and V′, P′, F′, and t' in another case. The principle of this rule will be best understood by reasoning out an example.

Imagine a vessel containing 1000 cubic feet of air at 0°, saturated with vapour, and having a barometer enclosed indicating say 32·18 inches of mercury as the pressure of the mixture of air and vapour. Now it is an axiom that the pressure of such a mixture is the sum of the separate pressures or elasticities of the air and the vapours; and as the elastic force of vapour at 0° is ·044, it follows that the pressure of the air alone is in our case $32 \cdot 18 - \cdot 044 = 32 \cdot 136$ inches. If the mixture be heated to say 112°, it will behave exactly as *dry* air or gas if no water be present to supply vapour; and the pressure remaining the same, the volume would become $1000 \times \frac{458 \cdot 4 + 112}{458 \cdot 4 + 0°} = 1244$ cubic feet. If now a little water be added to saturate the heated air, the tension of vapour at 112° being 2·731, the pressure would in a closed vessel be increased to $32 \cdot 136 + 2 \cdot 731 = 34 \cdot 867$; and if the vessel be enlarged until that pressure be reduced to its normal state of 32·18 inches, its capacity would become $1244 \times \frac{34 \cdot 867}{32 \cdot 18} = 1348$ cubic feet. Putting this in the form of the formula already given, we have $1000 \times \frac{32 \cdot 136 + 2 \cdot 731}{32 \cdot 136 + \cdot 044} \times \frac{458 \cdot 4 + 112}{458 \cdot 4 + 0} = 1348$ cubic feet, as before. General Roy made some experiments on the expansion of moist air at a pressure of 32·18 inches. His results are given in Table 21. Col. 4 is calculated by the formula.

TABLE 21.—Of the EXPANSION of MOIST AIR, at a pressure of 32·18 in. of the Barometer, from the Experiments of GENERAL ROY.

Temperature.	Elastic force of Vapour, inches of Mercury.	Volume by Experiment.	Volume by Calculation.
0°	·044	1000·00	1000·0
32	·181	1071·29	1074·5
52	·388	1123·05	1125·1
72	·785	1182·50	1183·7
92	1·501	1255·14	1255·0
112	2·733	1353·75	1348·3
132	4·752	1491·06	1476·4
152	7·931	1689·00	1657·0
172	12·752	1929·78	1918·3
192	19·837	2287·44	2291·0
212	30·000	2672·00	2824·0

If the vapour present is other than that of water, of course the elastic force of that particular vapour must be taken, from Table 22.

TABLE 22.—Of the ELASTIC FORCE of the VAPOUR of ALCOHOL, ETHER, &c. &c., in inches of Mercury, from the Experiments of REGNAULT.

Temp. Fahr.	Water.	Alcohol.	Ether.	Oil of Turpentine.	Sulphuret of Carbon.	Chloroform.
– 4°	·036	·130	2·724	..	..	..
+14	·082	·256	4·457	..	3·110	..
32	·181	·500	7·177	·082	5·012	..
50	·361	·949	9·311	·091	7·846	5·133
68	·685	1·732	17·118	·169	11·740	7·488
86	1·242	3·087	25·079	·276	17·110	10·870
104	2·162	5·279	35·968	·441	24·311	14·331
122	3·621	8·673	49·921	·677	33·571	20·642
140	5·858	13·780	68·122	1·059	45·772	29·055
158	9·177	21·228	90·925	1·649	60·982	38·433
176	13·962	32·000	116·031	2·409	88·941	53·850
194	20·687	47·866	153·504	3·583	112·035	71·319
212	29·921	66·338	193·717	5·311	130·760	92·701
230	42·337	92·591	246·024	7.374	162·846	118·913
248	58·712	126·291	..	10·118	201·638	150·315
266	79·932	170·520	..	13·661	246·480	185·866
284	107·000	221·957	..	18·201	..	..
302	141·000	285·740	..	23·799	..	..

(18.) "*Thermometers.*"—The expansion of bodies by heat has afforded the most convenient method of measuring temperatures. In the common thermometers mercury is used for medium temperatures, say from 0° to 600°; for lower temperatures alcohol is used, because it always remains fluid, even with the greatest cold which can be produced by artificial means; for high temperatures metals are commonly used. There is an imperfection in all these bodies as measurers of heat, for as we have seen (13) their expansions are not equable for equal increments of heat; but in the case of the mercurial thermometer it most fortunately happens that the variations of expansion in glass and mercury almost exactly compensate each other, so that mercury in glass has an expansion nearly equable at medium temperatures. Table 23, calculated from the refined experiments of Regnault, gives the error of the mercurial thermometer in common glass, such as is ordinarily used for thermometers, and it shows that, for temperatures under 540°, the error is less than 1° Fahr.; above that temperature the error becomes rapidly greater, and amounts to 7·2° at 662°. The amount of error seems to vary greatly with the kind of glass employed: with fine crystal glass the error at 662° was found by Regnault to be as much as 19°.

TABLE 23.—Of the ERROR of the COMMON MERCURIAL THERMOMETER in GLASS TUBE, from the Experiments of REGNAULT.

Temperature by an Air Thermometer.	Error of Mercurial Thermometer in Degrees Fahr.	Temperature by an Air Thermometer.	Error of Mercurial Thermometer in Degrees Fahr.	Temperature by an Air Thermometer.	Error of Mercurial Thermometer in Degrees Fahr.
212°	·000	374°	−·630	536°	+ ·936
230	−·036	392	−·550	554	+1·440
248	−·090	410	−·450	572	+1·944
266	−·162	428	−·360	590	+2·610
284	−·270	446	−·270	608	+3·240
302	−·360	464	−·180	626	+4·320
320	−·468	482	+·090	644	+5·400
338	−·576	500	+·360	662	+7·200
356	−·666	518	+·684		

"*Air Thermometers.*"—Air offers the great advantage as a measurer of heat, that its expansions are equal for equal increments of heat, and it is particularly valuable at high temperatures, where the departures of solids from uniformity become very serious: this is strikingly shown by Tables 24 and 25, from

TABLE 24.—Of the MELTING POINT of METALS, &c., according to the Experiments of Pouillet; the temperatures above dull red heat were measured by an Air Thermometer.

	Fahr.	
Wrought Iron, English, hammered	2910°	M. Pouillet.
,, French, soft	2730	,,
Steel, maximum	2550	,,
,, minimum	2370	,,
Cast Iron, grey, 2nd fusion	2190	,,
,, ,, very fusible	2010	,,
,, white, maximum	2010	,,
,, ,, minimum	1920	,,
Gold, very pure	2280	,,
,, standard coin	2156	,,
Silver, very pure	1830	,,
Brass	1650	,,
Antimony	810	,,
Zinc	793	,,
Lead	630	,,
Bismuth	518	,,
Tin	455	,,
Sulphur	239	,,
Wax, white	154	,,
,, unbleached	143	,,
Spermaceti	120	,,
Stearine	109° to 120	,,
Phosphorus	109	,,
Tallow	92	,,
Oil of Turpentine	14	,,
Mercury	−40	,,
Bismuth, 4; Tin, 1; Lead 1	201	,,
,, 8 ,, 3 ,, 5	212	,,
,, 5 ,, 3 ,, 2	212	,,
,, 5 ,, 4 ,, 1	246	,,
,, 1 ,, 1 ,, 0	286	,,
,, 1 ,, 2 ,, 0	334	,,
,, 0 ,, 3 ,, 1	334	,,
,, 0 ,, 3 ,, 1	367	,,
,, 0 ,, 4 ,, 1	372	,,
,, 0 ,, 5 ,, 1	381	,,
,, 0 ,, 2 ,, 1	385	,,
,, 1 ,, 3 ,, 0	392	,,
,, 0 ,, 1 ,, 1	466	,,
,, 0 ,, 1 ,, 3	504	,,
Common Salt, 1; Water, 3	4	Dr. Ure
Sulphuric Acid, sp. gr., 1·6415	−45	,,
,, Ether	−46	,,
Ice	32	,,

TABLE 25.—TEMPERATURES corresponding to various DEGREES of LIGHT in HEATED METAL, FURNACES, &c. M. POUILLET.

	Fahr.
	°
Red, just visible	977
,, dull	1290
,, cherry, dull	1470
,, ,, full	1650
,, ,, clear	1830
Orange, deep	2010
,, clear	2190
White heat	2370
,, bright	2550
,, dazzling	2730

the experiments of Pouillet and others with air thermometers, which exhibit glaring errors in the old tables of Wedgwood, Morveau, and Daniell.

The *scale* of the thermometer is arbitrary; in this country the scale of Fahrenheit is used, the distance between the freezing and boiling point of water being divided into 180 degrees, and Zero or 0° is fixed 32° below the freezing-point. On the continent the Centigrade scale is commonly used: here 0° is the freezing-point of water, and 100° the boiling-point. The readings of these two scales are easily convertible by the rules $(C \times 1{\cdot}8) + 32 = F$ and $\frac{(F - 32)}{1{\cdot}8} = C$, in which F and C represent degrees in the respective scales, Fahrenheit and Centigrade. Plate 12. gives a direct comparison of these scales to all useful temperatures.

TABLE 26.—Of the MEAN TEMPERATURE of the AIR, in various parts of the Globe, at different Seasons of the Year.

	Height above the Sea in Feet.	Mean Temperature.				
		Year.	Spring.	Summer.	Autumn.	Winter.
Irkutsk, Siberia	..	14°·5	17°·1	63°·0	20°·1	−38°
Nain, Labrador	..	25·5	21·6	45·7	36·0	− 1·3
St. Bernard, Alps	15890	30·2	28·4	43·0	31·3	18·0
St. Gothard, Alps	6873	30·6	27·1	44·1	32·0	18·3
Petersburgh	..	38·3	35·1	60·3	40·5	16·7
Moscow	480	38·5	43·3	62·6	34·9	13·5

TABLE 26—*continued.*

	Height above the Sea in Feet.	Mean Temperature.				
		Year.	Spring.	Summer.	Autumn.	Winter.
Christiana	..	41°·7	39°·2	59°·5	42°·4	25°·2
Stockholm	134	42·1	38·3	61·0	43·7	25·5
Montreal	..	43·7	44·2	69·1	47·1	17·5
Warsaw	397	45·5	44·6	63·5	46·4	27·5
Berne	1918	46·0	45·8	60·4	47·3	30·4
Stromness (Orkney) ..	..	46·4	43·7	54·5	48·2	39·2
Copenhagen	..	46·8	43·7	63·0	48·7	31·3
Dresden	397	47·3	47·1	62·9	47·1	31·3
Edinburgh	288	47·5	45·7	57·9	48·0	38·5
Berlin	128	47·5	46·4	63·1	47·8	30·6
Nicolaief..	..	48·7	49·3	71·2	50·0	25·9
Vienna	512	50·2	50·9	68·5	50·9	33·4
London	..	50·7	49·1	62·8	51·3	39·6
Paris	210	51·4	50·5	64·6	52·2	37·9
Penzance	..	52·0	49·8	61·7	53·8	43·9
Hobart Town	..	52·3	52·9	63·1	51·6	42·1
Turin	915	53·1	53·1	71·6	53·8	33·4
Trieste	288	55·8	53·8	71·5	56·7	39·4
Constantinople	..	56·7	51·8	73·4	60·4	40·6
Montpellier	..	57·4	56·8	75·9	61·0	44·5
Madrid	2175	57·6	57·6	74·1	56·7	42·1
Rome	174	59·7	57·4	73·2	61·7	46·6
Nice	..	60·1	55·9	72·5	63·0	48·7
Quito	9560	60·1	60·3	60·1	63·5	59·7
Naples	180	61·5	59·4	74·8	62·2	49·6
Lisbon	236	61·5	59·9	71·1	62·6	52·3
Buenos Ayres	..	62·5	59·4	73·0	64·6	52·5
Palermo	180	63·0	59·0	74·3	66·2	52·5
Algiers	..	64·0	63·0	74·5	70·5	54·0
Paramatta (Australia) ..	..	64·6	66·6	73·9	64·8	54·5
Madeira (Funchal) ..	..	65·7	63·5	70·0	67·6	61·3
Cape of Good Hope ..	..	66·4	65·5	74·1	66·9	58·6
Canton	..	69·8	69·8	82·0	72·9	54·8
Cairo	..	72·3	71·6	84·6	74·3	58·5
Ceylon (Candy)	1683	72·9	74·3	73·0	72·3	72·1
Rio Janeiro	..	73·6	72·5	79·0	74·5	68·5
Calcutta	..	78·4	82·6	83·3	80·0	67·8
Jamaica	..	79·0	78·3	81·3	80·0	76·3

NOTE.—The Seasons in this Table are (for the Northern Hemisphere): Spring—March, April, and May; Summer—June, July, and August; Autumn—September, October, and November; and Winter—December, January, and February.

(19.) The law of Marriotte (15) is true only so long as the temperature remains constant; but every change of pressure, and its accompanying change of volume, is simultaneously accompanied by a change of temperature, compression causing increase, and dilatation decrease. Thus if we take air at 60° and suddenly compress it to one-fourth of its volume as in (15), we should, by the law of Marriotte, have increased the pressure to $\frac{60}{15} = 4$ atmospheres; but we shall see presently that the compression causes an increase in temperature of 394° Fahr., as in col. 7 of Table 27, which will cause, in a closed vessel, a further increase of pressure, and instead of 4 atmospheres, we should have 7, as per col. 1, which, however, would subside to 4 when the temperature was reduced by the dissipation of heat to the normal temperature of 60°.

TABLE 27.—Of the HEAT produced by the COMPRESSION, and of the COLD produced by the DILATATION of AIR, the Volume at Atmospheric Pressure being 1·0 at the Temperature of 59° Fahrenheit.

Pressure				Volume.	Temperature of the Air throughout the Process.	Total Increase or Decrease of Temperature.
Above a Vacuum.			Above the Atmosphere in pounds per Sq. in.			
Atmospheres.	Inches of Mercury.	Pounds per Square in.				
					°	°
·167	5·	2·45	..	3·56	−151·6	−210·6
·333	10·	4·9	..	2·18	− 82·8	−141·8
·5	15·	7·35	..	1·634	− 35·7	− 94·7
·667	20·	9·8	..	1·333	+ 1·4	− 57·6
·833	25·	12·25	..	1·137	+ 32·0	− 27·0
1·000	30·	14·7	0·000	1·000	59·0	0·0
1·1	33·	16·17	1·47	·9346	73·58	14·58
1·25	37·5	18·37	3·67	·838	83·84	24·84
1·5	45·0	22·05	7·35	·726	105·08	46·08
1·75	52·5	25·81	11·11	·6724	150·62	91·62
2·0	60·	29·4	14·7	·6116	175·00	116·0
3·0	90·	44·1	29·4	·4588	255·	196·
4·0	120·	58·8	44·1	·374	317·	258·
5·0	150·	73·5	58·8	·320	369·	310·
6·0	180·	88·2	73·5	·2807	414·	355·
7·0	210·	102·9	88·2	·2515	453·	394·
8·0	240·	117·6	102·9	·2290	490·	431·
9·0	270·	132·3	117·6	·2062	524·	464·
10·0	300·	147·0	132·3	·1953	553·	494·
(1)	(2)	(3)	(4)	(5)	(6)	(7)

The general relations between volume, pressure, and temperature, are expressed by the following formulæ :—

$$T_2 = \left(\frac{P_2}{P_1}\right)^{\cdot 2908} \times (273 + T_1) - 273 ; \text{ and } \frac{V_2}{V_1} = \frac{P_1}{P_2} \times \frac{273 + T_2}{273 + T_1},$$

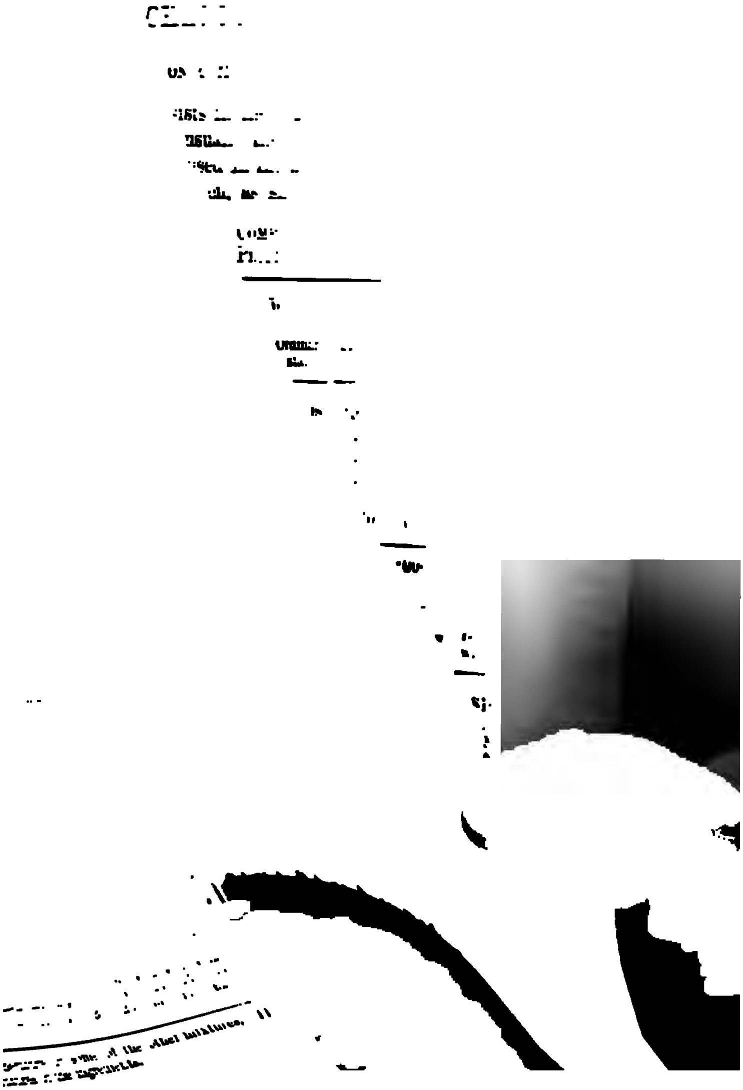
in which P_1 V_1 and T_1 are the pressure, volume, and temperature in Centigrade degrees in one case, and P_2 V_2 and T_2 in another case ; P_1 and P_2 are *total* pressures above a vacuum. Table 27 has been calculated by these rules.

(20.) The heat thus evolved by the compression of air is found very troublesome in some cases, such as where air-pumps are used to supply air to diving-bells, and pneumatic piles for bridges, &c., being destructive to the lubricating oil, and to the valves, &c. It is usual in such cases to surround the barrels with cold water, continually changed by a current passing through the cistern, and we can calculate the quantity of water necessary in any particular case. Say we take the case of a set of double-acting pumps, having two barrels, each 12 inches diameter and 18 inches stroke, which, working against 100 feet of water, with 35 revolutions per minute, and about 15 horse-power, would discharge 165 cubic feet of air, taken at the ordinary atmospheric pressure per minute. The weight of the air is 165 × ·0761 = 12·6 lbs., and as it is compressed with a pressure of four atmospheres, the heat generated will be 258° by col. 7 in Table 27, not 394°, as it would have been if the extra pressure generated by the temperature had not been

TABLE 28.—Of FRIGORIFIC MIXTURES for the ARTIFICIAL

	Proportional Parts, by												
Water	16	16	1	1	..	..	..	..	..	..	..	..	..
Sal Ammoniac	5	5	..	..	..	4	..	..	..	..	..	..	..
Nitre	5	5	..	..	..	2	..	..	..	..	..	..	..
Common Salt	..	..	..	..	..	..	..	..	..	..	..	1	..
Nitrate of Ammonia	..	..	1	1	..	..	5	..	6	..	..	..	3
Sulphate of Soda	..	8	..	..	3	6	6	..	..	8	5	..	..
Carbonate of Soda	..	..	..	1	..	..	..	..	..	..	..	..	..
Phosphate of Soda	..	..	..	..	..	..	..	9	9	..	..	..	5
Potash	..	..	..	..	..	..	..	..	..	..	..	..	..
Muriate of Lime	..	..	..	..	..	..	..	..	..	..	..	..	..
Crystallized Lime	..	..	..	..	..	..	..	..	..	..	..	..	..
Snow or pounded Ice	..	..	..	..	..	..	..	..	..	..	..	1	..
Muriatic Acid	..	..	..	..	..	..	..	..	..	5	..	..	..
Diluted Nitric Acid	..	..	..	..	2	4	4	4	4	..	..	..	4
Diluted Sulphuric Acid	..	..	..	..	..	..	..	..	..	..	4	..	..
Temperature of Ingredients	+50	+50	+50	+50	+50	+50	+50	+50	+50	+50	+50	+32	0
Temperature of the Mixture	+10	+4	+4	−7	−3	−10	−14	−12	−21	0°	+3	0	−34
Cold produced	40°	46°	46°	57	53	60	64	62	71	50	47	32	34

NOTE.—To obtain these results the temperature of the ingredients must be reduced previously temperatures given, whatever may be the



Cold may also be produced by the dilatation of air, as shown by col. 7 in Table 27. If compressed air be deprived by cold water or otherwise of the heat naturally developed by compression, and then suffered to return to its normal volume by the relief of the pressure, a very low temperature may be obtained. Cold is also produced by evaporation (196).

(22.) "*The Temperature of the Air at the Surface of the Globe.*"—The temperature of the air varies not only with geographical position, but also with the height above the sea level, and with local circumstances, so that observation alone can determine the mean temperature of any place. Table 26 gives the mean for the year, and for the four seasons, from the table of M. Mahlmann in Humboldt's great work on Central Asia, &c. Table 63 gives the mean temperature of every tenth day throughout the year in this country.

The influence of elevation above the sea is very considerable, but seems to vary with the climate, the season, and the contour of the ground. Where the slope is gradual, the cold produced is 1° Fahr. for about 430 feet; on steep mountain slopes about 355 feet; and in balloon ascents about 330 feet.

The temperature of the *surface* of the ground, follows pretty closely that of the air, but at a certain depth we come to a stratum whose temperature is invariable throughout the year, and is equal to the *mean* temperature of the air at that place. This depth varies, being in this country about 20 feet; in tropical America the depth is only 2 feet.

Below this stratum of uniform temperature the heat increases at about the rate of 1° Fahr. for every 58 feet of depth, so that if we assume 62° for the temperature of the surface, water would boil at $\frac{(212 - 62) \times 58}{5280} = 1{\cdot}647$ miles, and cast-iron would melt at $\frac{(2012 - 62) \times 58}{5280} = 21{\cdot}4$ miles.

CHAPTER II.

ON COMBUSTION.

(23.) Combustion consists in the combination of bodies with oxygen, the result being usually the development of heat and light. The combustibles used in the arts are principally composed of carbon and hydrogen, as shown by Table 29. The carbon, combining with oxygen derived from the air, forms carbonic acid, and the hydrogen similarly combining, forms water.

TABLE 29.—Of the CHEMICAL COMPOSITION of COMBUSTIBLES, according to PÉCLET, &c.

Elements.	Coal.	Coke.	Wood.			Peat.	
			Perfectly dry.	Ordinary state.	Char-coal.	Perfectly dry.	Ordinary state.
Carbon	·812	·850	·510	·408	·930	·580	·464
Hydrogen	·048	..	·053	·042	..	·060	·048
Oxygen	·054	..	·417	·334	..	·310	·248
Nitrogen and Sulphur	·031	..	..	..	..	..	..
Water	..	..	..	·200	..	..	·200
Ashes	·055	·150	·020	·016	·070	·050	·040
Total	1·000	1·000	1·000	1·000	1·000	1·000	1·000

Elements.	Oil of Turpentine.	Alcohol.	Olive Oil.	Sulphuric Ether.	Tallow.	Bees-wax.
Carbon	·884	·5198	·7721	·6531	·790	·816
Hydrogen	·116	·1370	·1336	·1333	·117	·139
Oxygen	..	·3432	·0943	·2136	·093	·045
Nitrogen and Sulphur	..	..	..	..	..	..
Water	..	..	..	..	..	..
Ashes	..	..	..	..	..	..
Total	1·000	1·000	1·000	1·000	1·000	1·000

Carbonic acid is composed of one equivalent of carbon (or 75), and two equivalents of oxygen (or 200), therefore a pound of carbonic acid consists of $\frac{75}{75+200} = \cdot 2727$ carbon, and $\frac{200}{75+200} =$ ·7273 oxygen.

Water is composed of one equivalent of oxygen (or 100), and one of hydrogen (or 12·5), therefore a pound of water consists of $\frac{12\cdot5}{100 + 12\cdot5} = \cdot111$ hydrogen, and $\frac{100}{100 + 12\cdot5} = \cdot889$ oxygen.

TABLE 30.—Of the CALORIFIC POWER of COMBUSTIBLES, by Experiment.

	Units of Heat per lb. of Fuel, by Experiment.	Authorities.	Units of Heat per lb. of Fuel, by Theory.
Hydrogen, burning to Water	62535	Dulong.	
,, ,, ...	62032	Favre & Silbermann.	
,, ,, ...	42120	Laplace.	
,, ,, ...	39807	Clement.	
,, ,, ...	42552	Despretz.	
Carbon, burning to Carbonic Acid..	12906	Dulong.	
,, from wood, burning to Carbonic Acid	14544	Favre & Silbermann.	
Carbon, burning to Carbonic Acid..	14040	Despretz.	
,, ,, Carbonic Oxyde	2495	Dulong.	
,, ,, ,,	4453	Favre & Silbermann.	
Carbonic Oxyde, burning to Carbonic Acid	4478	Dulong.	
,, ,, ,,	4325	Favre & Silbermann.	
1 lb. Carbon in the form of Carbonic Oxyde, burning to Carbonic Acid	10411	Dulong.	
Wood, perfectly dried by artificial heat	6480	Rumford	6582
Wood, in ordinary state of dryness..	5040	,, ...	5265
Olive Oil	17752	Dulong	17482
,, ...	16279	Rumford.	
,, ...	20153	Lavoisier.	
Colza Oil	16753	Rumford.	
Alcohol	12339	Dulong	12593
,, ...	11151	Rumford.	
Sulphuric Ether	16974	Dulong	15095
,, density .728 at 68° ..	14454	Rumford.	
Tallow	15550	,, ...	16792
,, ...	12935	Laplace.	
Oil of Turpentine	19505	Dulong	18663
Naphtha, density ·827	13208	Rumford.	
Sulphur	4682	Dulong.	
,, ...	4032	Favre & Silbermann.	
Phosphorus	13500	Laplace.	
Bees' Wax, white	18900	,, ...	18873
,, ,, ...	17422	Rumford.	

(24.) The experiments of Dulong, as given in Table 30, show that one pound of carbon, combining with the necessary quantity of oxygen, develops 12906 units of heat; and one pound of hydrogen, similarly combining, yields 62535 units. The unit of heat is the amount necessary to heat one pound of water 1° Fahr. See (1.)

(25.) When a combustible contains hydrogen and oxygen in the proportion required to form water, they combine during the process of combustion, but give out no useful heat. If hydrogen alone is present, it yields usefully the full amount of heat due to it. When oxygen is present, but in too small a proportion to combine with the *whole* of the hydrogen, there remains an *excess* of hydrogen, which yields its due proportion of heat as before.

(26.) Taking as an illustration, first, a body containing no oxygen—say oil of turpentine, which is shown by Table 29 to be composed of ·884 carbon, and ·116 hydrogen—then one pound will give—

Carbon .. ·884 × 12906 = 11409 units of heat
Hydrogen .. ·116 × 62535 = 7254 „
18663 „

(27.) ALCOHOL will serve to illustrate the effect of oxygen in a combustible, it being composed, as per Table 29, of ·5198 carbon, ·137 hydrogen, and ·3432 oxygen. We have seen in (23) that oxygen requires one-eighth part of its weight of hydrogen to form water; the oxygen in alcohol will require $\frac{\cdot 3432}{8}$ = ·0429 hydrogen; whereas the combustible contains ·137 of hydrogen: there remains, therefore, ·137 − ·0429 = ·0941 hydrogen *in excess* to develop its heat, and we have—

Carbon .. .. .. ·5198 × 12906 = 6708 units
Hydrogen in excess .. ·0941 × 62535 = 5885 „
12593 „

Similarly for OLIVE OIL, composed of ·7721 carbon, ·1336 hydrogen, and ·0943 oxygen, we have—

Carbon .. .. ·7721 × 12906 = 9965 units
Hydrogen $\left(\cdot 1336 - \frac{\cdot 0943}{8}\right)$ = ·1218 × 62535 = 7517 „
17482 „

Tallow, composed of ·79 carbon, ·117 hydrogen, and ·093 oxygen, will give—

Carbon .. .. $\cdot79 \times 12906 = 10195$ units

Hydrogen $\left(\cdot117 - \frac{\cdot093}{8}\right) = \cdot1055 \times 62535 = 6597$ „

$\overline{16792}$ „

Sulphuric Ether, composed of carbon ·6531, hydrogen ·1333, and oxygen ·2136, will give—

Carbon .. .. $\cdot6531 \times 12906 = 8429$ units

Hydrogen $\left(\cdot1333 - \frac{\cdot2136}{8}\right) = \cdot1066 \times 62535 = 6666$ „

$\overline{15095}$ „

Beeswax, composed of carbon ·816, hydrogen ·139, and oxygen ·045, will give—

Carbon .. .. $\cdot816 \times 12906 = 10531$ units

Hydrogen $\left(\cdot139 - \frac{\cdot045}{8}\right) = \cdot1334 \times 62535 = 8342$ „

$\overline{18873}$ „

(28.) From the Government experiments on Coal, by Playfair and De la Bèche, we find, as a mean of 17 different kinds of English, Welsh, and Scotch coals, the composition to be as per Table 29, and from this we have—

Carbon .. .. $\cdot812 \times 12906 = 10480$ units

Hydrogen $\left(\cdot048 - \frac{\cdot054}{8}\right) = \cdot041 \times 62535 = 2564$ „

$\overline{13044}$ „

Coke, containing ·85 carbon, without any hydrogen or oxygen, will yield ·85 × 12906 = 10970 units of heat.

(29.) Wood, perfectly dry, contains not only ·51 carbon, but also ·47 hydrogen and oxygen, but as these are in the proportion proper for forming water, they combine without yielding any useful heat, and we have ·51 × 12906 = 6582 units per pound of dry wood.

Wood in its ordinary state of dryness contains 20 per cent. of water, its carbon is thereby reduced to ·51 × ·8 = ·408, and its calorific power to ·408 × 12906 = 5265 units per pound.

Charcoal from wood, containing ·93 carbon, will give ·93 × 12906 = 12002 units per pound.

(30.) Peat, artificially dried, contains ·58 carbon, ·06 hydrogen, and ·31 oxygen. The oxygen in the fuel will combine with $\frac{\cdot 31}{8}$ = ·04 hydrogen, leaving ·06 − ·04 = ·02 hydrogen in *excess*, to develop its share of heat; and we have—

Carbon .. .. ·58 × 12906 = 7485 units
Hydrogen in excess .. ·02 × 62535 = 1251 „
8736 „

Peat in its natural state of dryness contains ·464 carbon, ·048 hydrogen, and ·248 oxygen. The oxygen will combine with $\frac{\cdot 248}{8}$ = ·031 hydrogen, leaving ·048 − ·031 = ·017 hydrogen in excess, and we have—

Carbon .. .. .. ·464 × 12906 = 5988 units
Hydrogen in excess .. ·017 × 62535 = 1163 „
7151 „

Charcoal of peat contains ·818 carbon, and yields ·818 × 12906 = 10557 units per pound.

(31.) The heating power of combustibles, as found by the preceding calculations, is the maximum effect they are capable of producing. When we come to apply it to practice we shall see (96) that there are sources of unavoidable loss which reduce their useful effect considerably.

(32.) The heating power of combustibles may also be determined by direct experiment; this has been done by calorimeters specially designed for the purpose, such as Rumford's and others. But perhaps the most valuable experiments are those that have been made on the large scale on steam-engine boilers in practice; we then obtain the heating power of fuel, in a form directly applicable to cases analogous to the experimental ones.

(33.) Some care is necessary in applying experimental data to particular cases, to be certain that the circumstances are identical with the experimental ones. For instance, if we make experiments on a steam-boiler in full work, with steam up and fire in good order, and observe for some hours the quantity of fuel consumed, leaving off the experiment with steam up to the same pressure and fire in good order, &c., we get a certain result, but one that cannot be applied without correction to ordinary cases where a boiler is worked only 10 or 12 hours per day, and where there are losses during stoppages and at night, for which allowance must be made.

(34.) Rumford's apparatus consisted of a shallow vessel of copper filled with water; the fuel to be experimented on is burnt beneath it, and the products of combustion being collected by a hood or inverted funnel, are caused to pass by a worm circulating through the mass of the water, and the calorific power is estimated by the increase in temperature of the water and the vessel. To avoid loss of heat by the apparatus during the experiment, the *mean* temperature is arranged to be the same as the temperature of the air; thus if the air was at 60° and the range of temperature 20°, the water would be taken at 50° to begin with, and would be 70° at the end of the experiment. With such an apparatus we should not obtain the total heating power in the fuel; some of the heat would be lost by direct radiation from the burning fuel, and some would pass off with the air issuing from the apparatus after passing through it, which would always have a temperature higher than the water it was intended to heat.

(35.) Say we had an apparatus like Rumford's, made of sheet copper, weighing 10 pounds and holding 60 pounds of water at 50°, which became heated to 70° by $\frac{1}{4}$ of a pound or 4 ounces of dry wood. The specific heat of copper (Table 1) being ·095 by Regnault's experiments, the amount of heat absorbed by the vessel heated 20° will be $10 \times 20 \times \cdot 095 = 19$ units, and the water absorbs $60 \times 20 = 1200$ units. The $\frac{1}{4}$ lb. of fuel has thus given out $19 + 1200 = 1219$ units, which is equal to $\frac{1219}{\cdot 25} = 4876$ units per pound. Table 30 gives the calorific power of combustibles by Rumford and others.

(36.) "*Effect of Water in a Combustible.*"—When a combustible contains water, with which it is more or less saturated, the effect is twofold; say we take wood in its ordinary state, containing 20 per cent. of water; the really combustible matter is reduced to 80 per cent. of the amount contained in the same weight of wood perfectly dry, and of necessity the calorific power is reduced in the same proportion. The second effect is, that part of the heat in the residue is consumed uselessly in evaporating the water. Wood perfectly dry gives by Table 30, 6480 units, which for wood in the ordinary state containing 20 per cent. of water, is reduced to 6480 × ·8 = 5184 units, but the ·2 water, say at 62°, will require for its evaporation (1178 − 62) × ·2 = 223 units, so that the useful heat is reduced to 5184 − 223 = 4961 units: this correction may be neglected in many cases, the amount being small in proportion to the heat given out by the fuel.

(37.) The experiments of Rumford have shown that the heating power of wood varies only with the state of dryness; that is to say, all the different kinds of wood in the same state of dryness yield sensibly the same amount of heat.

(38.) The best experiments on the heating power of wood, on the large scale, are given by Péclet. In one case with a hot-water boiler in a public bath, 15,800 lbs. of water were heated 153° by 440 lbs. of wood in the ordinary state of dryness, containing 20 per cent. of water; this gives $\frac{15800 \times 153}{440}$ = 5494 units per pound of wood. This is an exceptionally good result, arising no doubt from the fact that the apparatus was so well arranged that the smoke left it at the temperature of the atmosphere, or nearly so, the whole of the heat given out by the wood being thus utilized.

(39.) In another experiment on a steam-boiler, 3·24 lbs. of water were evaporated per pound of wood: the air passed into the chimney at 480°, and retained half of its oxygen, or 10 per cent. unconsumed (60). The temperature of the feed-water is not given, but assuming that it was 100°, the calorific power of the wood is 3·24 × (1178 − 100) = 3493 units per pound of fuel.

(40.) This is a low result, but it is what might be expected under the circumstances. The wood used contained 25 per cent. of water, therefore a pound of such wood contained only ·75 lb. of real combustible; the water would require ·25 × (1178 − 62°) = 279 units to evaporate it, and there would be a further loss of heat by the air in the chimney which departs highly heated. We shall see in (55) and (60) that *dry* wood requires 80·5 × 2 = 161 cubic feet of air when the oxygen is only half consumed, as in our case; therefore wood containing 25 per cent. of water requires 161 × ·75 = 120 cubic feet; or 120 × ·0761 = 9·1 lbs. of air, which, heated 420°, or say from 60 to 480, the temperature of the air in the chimney will carry off 9·1 × 420 × ·238 = 910 units. Adding these together, we obtain 3493 + 279 + 910 = 4682 units per pound of damp wood. Rumford's experiments in Table 30 give 5040 units, the difference arises no doubt from loss of heat by radiation from the boiler, &c., &c., which, as we have seen in (34) was avoided in the apparatus of Rumford.

The circumstances under which this experiment was made are analogous to most practical cases, and the heating power of wood in its ordinary state of dryness may be taken at about 3500 units per pound of fuel.

(41.) Experiments were made upon the *weekly* consumption of coals by two Elephant or French boilers, composed of a body of large diameter, connected by necks with three smaller ones about 20 inches diameter: the furnaces were Juckes' self-acting. The experiments were made with two kinds of coal, one a kind of Welsh coal, and the other small coal, or screenings from Yorkshire coals; each kind was experimented on for 6 days, from Monday to Saturday, and the result includes, therefore, loss by radiation during the night, &c., &c., and in getting up steam each morning. The water evaporated was *measured* in a vessel from which the feed pump was supplied.

(42.) With the Welsh coals 193876 lbs of water were evaporated by 28160 lbs. of coals, or $\frac{193876}{28160} = 6 \cdot 88$ lbs. of water per pound of coal. The water had a temperature of about 80°, and hence we have $\frac{(1178 - 80) \times 193876}{28160} = 7560$ units per pound of Welsh coals.

(43.) With the Yorkshire small coals, 167526 lbs. of water were evaporated by 29802 lbs. of coal, or $\frac{167526}{29802} = 5 \cdot 621$ lbs. of water per pound of coal, or $\frac{(1178 - 80) \times 167526}{29802} = 6172$ units of heat per pound of Yorkshire small coal.

(44.) Both these results are low, as might be expected, but they apply without correction to analogous cases, which are very numerous. The engine *worked* about 12 hours per day.

(45.) There is a source of error in such experiments, frequently overlooked, arising from *priming*, in which water passes over with the steam, and is not evaporated to steam at all. In the two following experiments this was avoided by taking off the man-hole and allowing the steam to escape into the atmosphere; the water was measured in by hand with a two-gallon spirit measure.

Experiment with Carter's patent tubular boiler, in which the fire is placed beneath the body, passes along the bottom, returning to the front by a set of 4-inch tubes, and again to the back by another set of 3-inch tubes, thus traversing the length of the boiler thrice; the temperature of the air as it left the boiler was 330°. In six hours, 628 lbs. of Duffryn Welsh coals evaporated 5140 lbs. of water at 52° to steam at 212°, and we have therefore $\frac{(1178 - 52) \times 5140}{628} = 9216$ units per pound of coal.

(46.) This boiler was full 30-horse power, but could not be worked up to more than 10-horse in the experiment, because it primed or boiled over at the man-hole, when the fire was kept up to its proper intensity. This is what might have been expected; by Table 11, each pound of water evaporated, formed 1640 cubic feet of steam, whereas with say 45 lbs. pressure, only 439 cubic feet would have been formed, and the tendency to boil over would be proportional to the volume of steam to be extricated from the water in a given time. To produce the same amount of priming with 45 lbs. steam, this boiler would require to be worked up to $10 \times \frac{1640}{439} = 37$-horse power. The result was, that the fire-grate could not be kept covered with fuel, and the economic result was not as good as it might have been if the boiler could have been worked up to its full power.

(47.) A similar experiment was made with a Cornish boiler, the fire being inside as usual. In six hours, 361 lbs. of Duffryn coals evaporated 2600 lbs. of water at 58° to steam at 212°, and we have therefore $\frac{(1178 - 58) \times 2600}{361} \times 8066$ units per pound of coal.

(48.) In the extensive experiments made by H.M. Government, a Cornish boiler was used; the results obtained are given with others in Table 31, which gives a general view of the preceding experimental results; we have added in the third column the quantity of fuel required per horse-power, taking that to be equal to a cubic foot of water evaporated to steam, which is a good general practical rule. The quantity of heat required to raise the temperature of water from 60°, and to evaporate it is by (8) equal to $1178 - 60° = 1118$ units per pound, or $1118 \times 62 \cdot 32 = 69674$ units per cubic foot; thus in the case of Newcastle coal, we have $\frac{69674}{8230} = 8 \cdot 46$ lbs. of coal per cubic foot of water, as per Table 31, and it is the same at all temperatures.

TABLE 31.—Of the HEATING POWER of COMBUSTIBLES from cases in practice.

Kind of Boiler, &c.	Authorities.	Kind of Fuel.	Units of Heat or Pounds of Water heated 1° by 1 lb. of Fuel.	Pounds of Water at 212° to Steam at any Pressure per Pound of Fuel.	Pounds of Fuel to evaporate 1 Cubic foot of Water at 60°, to Steam.
Hot-water boiler ..	Péclet	Wood (·2 water)	5494	5·687	12·68
Steam-boiler	„	Wood (·25 water)	3500	3·62	19·90
Steam, Elephant, boiler (weekly result) ..	Easton & Amos	Welsh Coals ..	7560	7·62	9·21
„ „ „	„ „	Yorkshire, small	6172	6·40	11·28
Steam-boiler, Cater's Patent Tubular ..	„ „	Duffryn Coals ..	9220	9·52	7·55
Steam-boiler, Cornish single flued	„ „	„ „	8066	8·35	8·64
„ „ „	H. M. Commissioners	Welsh Coals ..	9080	9·40	7·67
„ „ „	„ „	Newcastle Coals	8230	8·52	8·46
„ „ „	„ „	Lancashire Coals	7612	7·88	9·15
„ „ „	„ „	Scotch Coals ..	7535	7·80	9·24
Steam-boiler	James Watt ..	Staffordshire Coals	6500	6·72	10·70
„ „	Tredgold ..	Peat (Dartmoor)	2400	2·48	23·00

(49.) "*Air required to support Combustion.*"—A knowledge of the quantity of air necessary for different combustibles is important, in order to determine the sizes of flues, &c.

(50.) We have seen in (23) that carbonic acid is composed of ·2727 carbon, and ·7273 oxygen; and as atmospheric air is composed of two equivalents (or 350) of nitrogen; and one equivalent (or 100) of oxygen, it follows that one pound of air consists of $\frac{100}{350 + 100} = \cdot222$ lbs. of oxygen, and $\frac{350}{350 + 100} = \cdot778$ lbs. of nitrogen. A pound of carbon will require $\frac{\cdot7273}{\cdot2727} = 2\cdot67$ lbs. of oxygen, which is the amount contained in $\frac{2\ 67}{\cdot222} = 12\cdot03$ lbs. of air; and as a cubic foot of air at 62° weighs, by Table 19, ·0761 lbs., this is equal to $\frac{12\cdot03}{\cdot0761} = 158$ cubic feet of common air at ordinary temperatures. This is the *minimum* amount necessary for the combustion of a pound of carbon.

(51.) Water being composed of ·111 hydrogen and ·889 oxygen, one pound of hydrogen requires $\frac{\cdot 889}{\cdot 111}$ = 8 lbs. of oxygen, which is the amount contained in $\frac{8}{\cdot 222}$ = 36 lbs., or $\frac{36}{\cdot 0761}$ = 473 cubic feet of common air at 62°, and this is the minimum amount necessary for the combustion of a pound of hydrogen.

(52.) From these elements we can easily calculate the quantity of air required for any combustible whose composition is known.

(53.) The average composition of coal is given by Table 29 at ·812 carbon, and ·048 hydrogen, which last is, by (28), reduced to ·041 hydrogen *in excess*, and we shall require (812 × 158) + (·041 × 473) = 147·6 cubic feet of air at 62°.

(54.) Coke, by Table 29, contains ·85 carbon, and will require ·85 × 158 = 134·3 cubic feet of air at 62° per pound.

(55.) Wood perfectly dried by artificial means, contains ·51 carbon, and will require ·51 × 158 = 80·5 cubic feet of air at 62° per pound.

(56.) Wood in its ordinary state of dryness contains, by Table 29, about 20 per cent. of water, and the carbon is thereby reduced to ·408 per pound of wood, which will require ·408 × 158 = 64·5 cubic feet of air per pound.

(57.) Charcoal, of wood, contains ·93 carbon, which will require ·93 × 158 = 147 cubic feet of air at 62° per pound.

(58.) Peat, perfectly dried by artificial means, contains ·58 carbon, ·06 hydrogen, and ·31 oxygen. The oxygen will combine with $\frac{1}{8}$th of its weight of hydrogen, or $\frac{\cdot 31}{8}$ = ·04, leaving ·06 − ·04 = ·02 hydrogen *in excess*, and the air required will be (·58 × 158) + (·02 × 473) = 101·1 cubic feet of air at 62° per pound of dry peat.

(59.) Peat, in its ordinary state of dryness, produced by long exposure to the air, contains 20 per cent. of water, and its carbon is reduced to ·464, the hydrogen to ·048, and oxygen to ·248. The oxygen will combine with $\frac{1}{8}$th of its own weight of hydrogen, or $\frac{\cdot 248}{8}$ = ·031, leaving thus ·048 − ·031 = ·017 hydrogen *in excess*, and the air required will be (·464 × 158)

+ (·017 × 473) = 81·3 cubic feet of air at 62° per pound of ordinary peat.

(60.) The quantities of air, as found by the preceding calculations, is the *minimum* absolutely necessary to furnish the oxygen required to support the combustion. Practice has led to the use of a much larger quantity of air than the theory indicates, the reason being, no doubt, to avoid the formation of carbonic *oxyde* instead of carbonic *acid*, which would be the result if the supply of oxygen were too small, see (108). Analyses of the air that has passed through the fires of well-arranged steam-boilers, &c., show that the air retains 10 per cent. of oxygen unconsumed, so that in such cases *double* the quantity of air has been used; and we may admit as a practical rule, that the quantity of air used should be *double* the minimum theoretical quantity. The effect of varying the quantity of air on the useful result will be seen by (101).

THE VOLUME OF GAS, &C., PRODUCED BY THE DIFFERENT COMBUSTIBLES.

(61.) If combustibles were formed of pure carbon, the only product would be carbonic acid: this is the case with some kinds of fuel, such as charcoal and coke. When a combustible contains hydrogen, it combines with 8 times its own weight of oxygen, derived from the oxygen in the fuel, or from the air which supports the combustion; in either case water is formed, which again becomes vapour.

(62.) When a combustible contains water already formed, and with which it is more or less saturated, vapour is formed from it, and is added to the products of combustion.

By (col. 5) in Table 32 we find that a pound of vapour of water at ordinary atmospheric temperature and pressure has a volume of 21·07 cubic feet. Carbonic acid gas occupies 8·59 cubic feet, and oxygen 11·88 cubic feet. When oxygen and carbon combine, the volume of the carbonic acid gas is nearly the same as that of the oxygen from which it is formed; we have seen (23) that a pound of carbonic acid gas contains ·7273 lb. of oxygen, which will have a volume of ·7273 × 11·88 = 8·64 cubic feet, being nearly the same as the volume of the carbonic acid gas formed, which, as we have seen, is 8·59 cubic feet; so that when a combustible contains carbon only, the volume of gas in the chimney is the same as the volume of air entering

TABLE 32.—Of the DENSITY of GASES and VAPOURS, Air at the same Temperature and Pressure being 1·0; also the Weight of a Cubic foot at 62°, under an Atmospheric Pressure of 29·92 inches of Mercury.

	Density, Air at the same Temp. and Pressure being 1 (Regnault).	Specific Gravity or Density, Water at 62° being 1·0.		Weight of a Cubic foot in Pounds.	Cubic feet at 62° in 1 lb.
Air (atmospheric)	1·00000	·001221	or $\frac{1}{819}$	·07610	13·14
Hydrogen Gas	·06926	·0000846	,, $\frac{1}{11820}$	·00527	189·70
Oxygen Gas	1·10563	·001350	,, $\frac{1}{844}$	·08414	11·88
Nitrogen Gas	·97137	·001185	,, $\frac{1}{741}$	·07383	13·54
Carbonic Acid Gas	1·52901	·001870	,, $\frac{1}{535}$	·11636	8·59
Vapour of Water	·6235	·0007613	,, $\frac{1}{1325}$	·04745	21·07
,, Alcohol	1·589	·00194	,, $\frac{1}{515}$	·12092	8·27
,, Sulphuric Ether	2·586	·00316	,, $\frac{1}{316}$	·19680	5·08
,, Oil of Turpentine	4·760	·00581	,, $\frac{1}{172}$	·36224	2·76
,, Mercury	6·976	·00850	,, $\frac{1}{118}$	·52987	1·88
	(1)	(2)	(3)	(4)	(5)

NOTE.—The densities of the *vapours* in column (2), &c., are reduced from their real pressure and density by calculation.

the fire expanded to the volume due to the increased temperature, for the nitrogen in the air passes through the fire without alteration, and the oxygen consumed is replaced by the same volume of carbonic acid gas.

(63.) By Table 58 (col. 11) one pound of water forms 1135 cubic feet of vapour, but that is at its own normal pressure of ·556 inches of mercury: say we had a closed room or vessel containing 1135 cubic feet of *dry air* at 62° and at 30 barometer pressure, a pound of water would become vapour and just suffice to saturate it, the pressure increasing to $30 + \cdot556 = 30\cdot556$ inches of mercury. By enlarging the vessel, we can again reduce the pressure to 30, and the volume required for that is—$\frac{1135 \times 30\cdot556}{30\cdot} = 1156$ cubic feet. The volume of air has therefore been increased $1156 - 1135 = 21$ cubic feet by the addition of one pound of water converted to vapour; and if the volume of air had been greater, the *increase* in volume would have been the same; thus if the volume of dry air had been ten times greater, or 11350 cubic feet, the pound of water would have filled it with vapour, but the tension of that vapour would have been $\frac{1}{10}$ only of the maximum tension, or ·0556 inch of mercury, and the volume at 30 would become—

$$\frac{11350 \times 30\cdot0556}{30} = 11371 \text{ cubic feet,}$$

being an increase of 11371 − 11350 = 21 cubic feet as before.

(64.) In most cases the temperature of the air in the chimneys of steam-boilers is about 550° (93), at which the volume of air by Table 19 is double the volume at 62°; and allowing, as in (60), that the oxygen should be only half consumed, and a double quantity allowed for good combustion, we have the following results:—

(65.) Coke, by (54), requires 134·3 cubic feet of air theoretically, which becomes 134·3 × 2 × 2 = 537 cubic feet in the chimney at 550°.

(66.) Charcoal gives 147 × 2 × 2 = 588 cubic feet at 550°. Coke and charcoal contain only carbon, as per Table 29, and the product of combustion is carbonic acid gas only.

(67.) Coals contain, besides ·812 carbon, ·048 of hydrogen, which will combine with ·048 × 8 = ·384 of oxygen (of which ·054 is derived from the combustible itself, and the rest from the air passing through the fire), producing ·048 + ·384 = ·432 water, which again becomes ·432 × 21·07 = 9·1 cubic feet of vapour; and we have altogether (147·6 × 2) + (9·1) × 2 = 608·6 cubic feet in the chimney, at 550° per pound of coal.

(68.) Wood perfectly dry contains ·47 of hydrogen and oxygen in the proper proportion for forming water, which becomes ·47 × 21·07 = 9·9 cubic feet of vapour, and we have (80·5 × 2) + (9·9) × 2 = 341·8 cubic feet of air, &c., at 550°.

(69.) Wood in the ordinary state of dryness contains ·20 water, and hydrogen and oxygen forming during combustion ·376 water more; and the two together will form (·20 + ·376) × 21·07 = 12·1 cubic feet of vapour; and we have (64·5 × 2) + (12·1) × 2 = 282·2 cubic feet of air at 550° per lb.

(70.) Peat, perfectly dry, contains ·06 hydrogen, which, combining with ·06 × 8 = ·48 oxygen, will form ·06 + ·48 = ·54 water, which again becomes ·54 × 21·07 = 11·3 cubic feet of vapour; and we have (101·1 × 2) + (11·3) × 2 = 427 cubic feet of air at 550°.

(71.) Peat in the ordinary state of dryness contains ·048 of hydrogen, which, combining with ·048 × 8 = ·384 of oxygen, will form ·048 + ·384 = ·432 water, which, added to the ·20 in the fuel, = ·432 + ·20 = ·632 water, becomes ·632 × 20·07 = 13·3 cubic feet of vapour; and we have (81·3 × 2) + (13·3) × 2 = 352 cubic feet of air and vapour at 550°.

ON THE MODES IN WHICH COMBUSTIBLES YIELD THEIR HEAT UNDER DIFFERENT CIRCUMSTANCES, &C.

(72.) To illustrate the way in which combustibles give out their heat, we will take the case shown by Fig. 1 of a ball heated to say 600°, and placed in a room or vessel whose walls are maintained at 100°, and let the space enclosed be a vacuum. Then the ball will give out its heat to the walls entirely by *radiation*, and will continue to do so till it is reduced to the same temperature as those walls.

(73.) Now take another case, represented by Fig. 2, in which a ball heated to 600° is placed in a room, whose walls have the same temperature as itself; and let air at say 100° be admitted at A and allowed to escape at B, becoming heated in its passage by contact with the ball and the walls, and cooling them continuously and simultaneously, until both are reduced to its own temperature. Here we have a case in which all the heat of the ball is given out *by contact of cold air*, and none by radiation, because the walls have throughout the same temperature as the ball.

(74.) To complete our illustrations, let Fig. 3 represent a case in which the heat of the ball is carried off by *both* causes. Say the ball is heated to 600°, while the walls are at 300° and the air at 100°; here part of the heat will be given out by radiation to the walls, as in our first illustration, and part to the cold air passing through the room, as in our second illustration.

(75.) Applying all this to combustibles, we find the case shown by Fig. 1 and (72) impossible, inasmuch as air is absolutely necessary to support combustion; but the other two cases are practicable.

(76.) Let Fig. 4 be a furnace constructed of fire-brick, which for the purpose of illustration we may suppose to be a perfect non-conductor of heat, so that its interior surface will have the same temperature as the fire, and we have a case analogous to Fig. 2 (73); no heat will be given out by radiation, but the whole of the caloric yielded by the fuel will be carried off by the air supporting the combustion. This air will be heated to a temperature varying with the volume of air admitted and the heating power of the fuel.

TABLE 33.—Of the HEATING POWER of COMBUSTIBLES, the quantity of Air required for the Combustion of One Pound of Fuel, and the Quantity and Temperature of the Air, &c., in the Chimney.

KIND OF FUEL.	Total Heat in a Pound.	Heat radiated.		Heat given out to Air supporting Combustion.		Air used.		Cubic feet of Vapour at 62°.	Cubic feet, Air and Vapour in Chimney at 550°.	Temp. of Air as it leaves the Fire.	
	Units.	Per cent.	Units.	Per cent.	Units.	lbs.	Cubic feet at 62°.			Steam-boiler with internal Fire.	Fire-brick Furnace.
Coals	13000	50	6500	50	6500	22·44	295	9·1	608·6	° 1279	° 2252
Coke	10970	50	5485	50	5485	20·47	269	..	537	1188	2089
Wood, artificially dried	6582	23	1514	77	5068	12·25	161	9·9	341·8	1869	2173
Wood, ordinary state (·2 water) ..	5265	23	1211	77	4054	9·817	129	12·1	282·2	1797	2091
Wood, Charcoal	12000	45	5400	55	6600	22·37	294	..	588	1326	2130
Peat, artificially, dried	8736	50	4368	50	4368	15·376	202	11·3	427	1256	2211
Peat, ordinary state (·2 water) ..	7151	50	3575	50	3575	12·4	163	13·3	352	1274	2244
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)

(77.) Supposing that the fuel used was coal, one pound of coal yields 13,000 units of heat (28), and requires 147·6 cubic feet of air (53) at 62°; and allowing double, as per (60), we have 147·6 × 2 = 295 cubic feet, or 22·44 lbs. of air per pound of coal. Thus a pound of coal will heat a pound of water 13000 degrees; and as the specific heat of air by Regnault's experiments is ·238, an equal weight of water being 1, it follows that a pound of coal will heat a pound of air $\frac{13000}{\cdot 238} = 54600^\circ$; and as we have 22·44 pounds of air to carry off this heat, the increase of temperature will be $\frac{54600}{22 \cdot 44} = 2434^\circ$, and as the air enters the fire at 62°, it must depart at $62^\circ + 2434^\circ = 2496^\circ$.

(78.) We assumed, for the purpose of illustration, that the fire-brick of which the furnace was constructed was a perfect non-conductor, which is not a fact—a considerable portion of the heat will be transmitted to the outer surface, and dissipated by radiation there.

Allowing that 10 per cent. of the heat is dissipated by radiation, &c., in the case Fig. 4, the heat carried off by the air is reduced to $13000 \times \cdot 9 = 11700$ units, and the temperature of the air to $(2434^\circ \times \cdot 9) + 62 = 2252^\circ$. The other kinds of fuel, yielding their respective quantities of heat to the air required for their combustion, will give different temperatures to that air as per (col. 11), Table 33, allowing throughout 10 per cent. for loss by radiation.

(79.) The temperature of the air will also vary with the volume admitted: in the case we have investigated the oxygen was half consumed. Table 34 gives the variation of temperature for coal with different volumes of air.

TABLE 34.—Of the TEMPERATURE of the AIR from a FIRE-BRICK FURNACE, showing the effect of using different volumes of Air.

State of the Oxygen.	Cubic feet.	lbs.	Increase of Temp.	Temp. of Atmosphere.	Temp. of Air as it leaves the Fire.
Half burnt ..	295	= 22·44	2190°	+ 62°	= 2252°
Quarter burnt ..	590	= 44·88	1095	+ 62	= 1157
One-fifth burnt ..	737	= 56·10	876	+ 62	= 938

(80.) It should be observed that the quantity of air cannot be reduced below a certain amount, for obviously the temperature of the air cannot be higher than the temperature of the fire itself. Now the temperature of a fire at white heat, according to Pouillet's Table 25, is 2370°; a cubic foot of air heated from 62° to 2370° requires $\cdot 0761 \times (2370 - 62) \times \cdot 238 = 41 \cdot 8$ units of heat; and the least quantity of air to carry off 11700 units is $\frac{11700}{41 \cdot 8} = 280$ cubic feet.

(81.) We will now investigate the phenomena of combustion in cases analogous to Fig. 3, where the fuel is more or less surrounded by surfaces of low temperature which absorb the radiant heat. By studying our three illustrations, it will be evident that the amount radiated will vary with the temperature of the absorbing surface, for if, as in Fig. 2, the walls were of the same temperature as the fire, no radiant heat would be given out or received, and generally the lower the temperature of the absorbent, the more radiant heat would be received by it.

THE RADIATING POWER OF COMBUSTIBLES.

(82.) Péclet's apparatus for ascertaining the radiating power of combustibles is shown in Fig. 5: it consisted of an annular vessel of tin plate, the interval between the two cylinders being filled with water, the temperature of which was given by two thermometers with long bulbs, whose stems pass through corks. The internal cylinder was open at both ends, its surface was coated with lamp-black, and in its centre was suspended a cage of wire containing the fuel. In using this apparatus a given weight of fuel in a state of ignition is introduced and consumed, the radiant heat and that alone is absorbed, and raises the temperature of the water and the vessel itself: knowing the weight of the water, and of the vessel, also the increase of temperature with a given weight of fuel, we can calculate the heat given out by radiation. But a correction is necessary here, for evidently with this form of apparatus we only obtain *part* of the heat radiated, because radiation takes place equally in *all directions*, and part of it must escape by the open ends of the cylinder, and be lost. In Péclet's apparatus the internal cylinder was 8 inches diameter and 12 inches high, and the ratio of the total radiant heat to the portion absorbed was in that case as 1·2 to 1, as shown by the following investigation.

(83.) The principles on which this correction is made are illustrated by Fig. 6, in which A is the radiant body, say at 600°, placed in the centre of a hollow sphere, C D E F, at 100°. The body, A, will send out radiant heat equally in all directions, which will be absorbed by the sphere, but if two segments, C D and E F, be cut out and removed, the radiant heat that would have fallen on them passes out into the atmosphere and is lost; but knowing the proportion which those two segments bear to the whole sphere, we can estimate the amount lost by their removal. The rules of mensuration show that the surface of any sphere or segment of a sphere, is given by multiplying the circumference of the sphere by the diameter, or by the height of the segment. The distances C E and C D are given, in our case 12 in. and 8 in. respectively, and the angle C E F being a right angle, we get the diameter C F of the circumscribing sphere $= \sqrt{12^2 + 8^2} = 14 \cdot 4$ inches or 45·2 inches circumference, and $45 \cdot 2 \times 14 \cdot 4 = 650$ total surface. The area of the two segments is $45 \cdot 2 \times 1 \cdot 2 \times 2 = 108$, leaving in the apparatus $650 - 108 = 542$ square inches to absorb the heat, the ratio of which to the surface of the whole sphere is $\frac{650}{542} = 1 \cdot 2$ to 1.

(84.) With this apparatus the weight of water was 23·84 lbs., and the vessel itself which was made of tin plate weighed 4·9 lbs. The combustion of ·1232 lbs. of wood charcoal raised the temperature 25°·2, the water received $23 \cdot 84 \times 25 \cdot 2 = 600 \cdot 7$ units of heat, and the vessel whose specific heat (2) was ·11 received $4 \cdot 9 \times 25 \cdot 2 \times \cdot 11 = 13 \cdot 6$ units, altogether $600 \cdot 7 + 13 \cdot 6 = 614 \cdot 3$ units, which with a completely surrounding surface would have been $614 \cdot 3 \times 1 \cdot 2 = 737$ units, or $\frac{737}{\cdot 1232} = 5983$ units per pound of fuel, and the total heating power being as we have shown (29) 12000 units, the radiant heat is $\frac{5983}{12000} = \cdot 5$ nearly, so that 50 per cent is given out by radiation, and 50 per cent to the air passing through the fuel.

(85.) The temperature of the ignited fuel would be about 2200°, and the *mean* temperature of the absorbing surface being 70° in the above experiment, the difference of temperature was $2200 - 70 = 2130°$, whereas if the absorbing surface had been at 300°, as in high pressure steam-boilers, the difference would have been $2200 - 300 = 1900°$ and the heat lost by radiation would have been $\frac{5983 \times 1900}{2130} = 5337$ units which is $\frac{5337}{12000} =$ ·445, or say 45 per cent of the heat in the fuel, leaving 55 per cent. to pass off in the air.

(86.) In another experiment with oak wood, made with the same apparatus, ·2145 lbs. of wood raised 23·84 lbs. of water and 4·9 lbs. of tin plate 9°, which is equal to

$$\frac{23 \cdot 84 + (4 \cdot 9 \times \cdot 11) \times 9 \times 1 \cdot 2}{\cdot 2145} = 1228 \text{ units per pound,}$$

and the total heating power of ordinary wood being (29) as we have shown 5265 units, the radiant heat is $\frac{1228}{5265} = \cdot 233$— say 23 per cent.; leaving 77 per cent. to pass off with the air.

(87.) In another experiment with peat charcoal, by the combustion of ·0858 lbs. of fuel the temperature was raised 15° 3, which is equal to $\frac{23 \cdot 84 + (4 \cdot 9 \times \cdot 11) \times 15 \cdot 3 \times 1 \cdot 2}{\cdot 0858} = 5217$ units per pound, and as the total heating power is by (30) 10557 units, the radiant heat is $\frac{5217}{10557} = \cdot 5$ nearly, or 50 per cent. of the total heat; leaving 50 per cent. to heat the air. Péclet estimates the radiant power of peat itself to be 50 per cent. also, the same as peat charcoal.

(88.) The radiant power of coal, coke, &c., could not be satisfactorily determined by this apparatus, from the difficulty of keeping a small quantity of fuel ignited; but by comparison Péclet estimates them as rather more than wood charcoal, which we found to be 45 per cent. (85); we may therefore estimate the radiant power of coal and coke at 50 per cent.

(89.) We have shown (77) that the air leaving a fire of coals, when *all* loss by radiation was avoided, had the temperature of 2496°; but when 10 per cent. was radiated the temperature was reduced to 2252°. When the fire is surrounded by cool surfaces 50 per cent. is radiated, and the temperature of the air is greatly reduced, the same weight having to receive only half the amount of heat. In the case of coals, the air leaves the fire at $\frac{13000 \times \cdot 5}{22\ 44 \times \cdot 238} + 62° = 1279°$; with wood in the ordinary state of dryness, $\frac{5265 \times .77}{9 \cdot 817 \times \cdot 238} + 62° = 1797°$; and so on with the rest.

Table 33 gives a general and collected statement of the facts arrived at in the foregoing investigations.

(90.) We may now apply all this to the case of a steam boiler, &c. We may assume that with ordinary firing about 5 per cent. more of the combustible matter in the fuel falls unconsumed from the grate, than in the experiments from which the data are derived, the useful heat being reduced to $13000 \times \cdot 95 = 12350$ units; we will also assume that each pound of coals requires 300 cubic feet, or 22·83 lbs. of air at 62°.

(91.) Let Fig. 7 be a boiler or fire-box—not of a practical form, but such as to serve for the illustration of the case in which the fire being completely surrounded by an absorbing surface, will *lose* none of its radiant heat. In such a case 50 per cent. of the total heat in the fuel will be given out by radiation, and 50 per cent. to the air by which the combustion is supported; the first, or 6175 units, is absorbed by the side, &c., of the fire-box, and 6175 units pass off with the air into the chimney and is lost. The temperature of the air as it departs may be found as before; the heat carried off by it would heat one pound of water 6175°, therefore a pound of air, $\frac{6175}{\cdot 238} = 25950^\circ$, and as we have 22·83 lbs. of air the increase of temperature will be $\frac{25950}{22 \cdot 83} = 1138^\circ$, and the final temperature $1138^\circ + 62^\circ = 1200^\circ$.

(92.) This temperature will vary with the quantity of air admitted, as is shown by Table 35.

TABLE 35.—Of the EFFECT of DIFFERENT VOLUMES of AIR in STEAM-BOILER FURNACES.

State of the Oxygen.	Cubic feet.	Pounds.	Increase of Temp.	Temp. of Atmosphere.	Temp. of Air as it leaves the Fire.
			°	°	°
Half burnt ..	300 =	22·83	1138 +	62 =	1200
Quarter burnt ..	600 =	45·66	568 +	62 =	630
One-fifth burnt..	750 =	57·00	455 +	62 =	517

(93.) We have seen that a boiler like Fig. 7, in which the air escapes immediately out of the furnace, 50 per cent. of the heat in the fuel is lost. Part of this may be recovered by causing the heated air to pass in contact with the surface of the boiler, through a long circuit, on its way to the chimney. In Fig. 8 we have a fire-box as in Fig. 7, and, for the purpose of illustration, we will suppose the boiler to have no return flue, but to be of great length, with the chimney at the end. It is found by observation of well-arranged boilers, that the air passes into the chimney at about 550°,—a high temperature is necessary to obtain the proper draught, as will appear hereafter (166); we have seen that as it leaves the fire-box it is at 1200°, while from end to end the absorbing surface in the case of a high-pressure boiler is at 300°.

(94.) The amount of heat abstracted from the air at each point will be in proportion to the *difference* between the temperature of the air at that point and the temperature of the boiler; at the fire-box end the difference is 1200° − 300° = 900°, while at the chimney end it is only 550° − 300° = 250°. Say we assume that one-tenth of this difference of temperature is parted with from point to point throughout, then the first difference being 900°, one-tenth of that is 90°, and the next point will have a temperature of 1200° − 90° = 1110°. The difference is now, therefore, 1110° − 300° = 810°, and the temperature of the second point will be $1110° - \frac{810}{10} = 1029°$, and thus we have calculated the successive temperatures in Fig. 8.

(95.) The air departing into the chimney at the high temperature of 552°, carries off a considerable amount of heat, namely the amount required to heat 22·83 pounds of air 490° (or 552° − 62°); and this is equal to 22·83 × 490 × ·238 = 2663 units, or about 20 per cent. of the total heat in the coals.

(96.) There is also another source of loss of heat, namely by radiation and contact of cold air with the outside of the boiler and its brickwork, whereby the boiler-house is kept at a high temperature. The loss from this cause will vary very greatly with the greater or less exposure, &c.; but for ordinary Cornish boilers, set in brickwork in the usual way in a closed boiler-house, we may take this loss at 12 per cent. Collecting these results we have the total heat in one pound of coal (13,000 units) distributed as shown by Table 36.

TABLE 36.—Of the DISTRIBUTION of the HEAT in ONE POUND of COALS, by an ordinary Boiler, with internal fire.

In Ashes, left unburnt	13000 × 5 % =	650	units.
Lost by Air in Chimney	13000 × 20 % =	2600	,,
Lost by Radiation, &c., in Boiler-house ..	13000 × 12 % =	1560	,,
Utilized in production of Steam	13000 × 63 % =	8190	,,
	Total	13000	,,

(97.) "*Efficiency of Long and Short Boilers.*"—We have stated (93) that a high temperature of the air in the chimney is necessary in order to obtain a good draught; but if it were not so, an extreme length of boiler would be necessary to secure even a small portion of the heat wasted. Say we doubled the length of our boiler, Fig. 8, continuing the calculation of the successive temperatures, we obtain the series given by Fig. 9. The heat lost by the chimney in this case would be the amount necessary to heat 22·83 lbs. of air 307° (or 369° – 62°), or 22·83 × 307 × ·238 = 1668 units. With the ordinary length of boiler the loss from the same cause was (95) 2663 units; so that by *doubling* the length we obtain 2663 – 1668 = 995 units only, or 12 per cent. increase on the useful effect (8190 units).

(98.) But even this would not be realized. In Table 36 we have allowed 1560 units as the loss in the boiler-house, &c., by a boiler of ordinary length: with double the length, the loss from the same cause would be greater, and would dissipate much of the extra heat obtained by the extra length. Indeed, it is obvious that beyond a certain point we should, by increasing the length, lose more from one cause than we should gain from the other. If we assume that in a boiler of ordinary length one half of the loss by radiation, &c., is due to the exposed front and furnace, and the other half to the body of the boiler, we shall be conducted to the results shown by Table 37.

TABLE 37.—Of the HEAT LOST by LONG and SHORT BOILERS, &c.

Proportional length of Boiler.	Temperature of Air in the Chimney	Heat lost by Air in Chimney.	Heat lost by Radiation, &c.			Left in the Ashes.	Heat utilized.	Ratio of Economy.	Gain or Loss per cent.
			By Front, &c.	By Body.	Total.				
	°	Units.	Units.	Units.	Units.	Units.	Units.		
½	778	3890	780 +	390 =	1170	650	7290	897	–10·3
¾	648	3184	780 +	585 =	1365	650	7802	960	– 4·0
1	552	2663	780 +	780 =	1560	650	8127	1000	0·0
1½	432	2010	780 +	1170 =	1950	650	8390	1032	+ 3·2
2	369	1668	780 +	1560 =	2340	650	8342	1026	+ 2·6

This would show that by reducing the boiler to half the usual length we should only lose 10 per cent.; one, half as long again as usual, would give only 3·2 per cent. more useful heat; and by increasing the length to double we should actually *lose* rather than gain. These results are not given as absolutely correct, but will at least serve to show approximately the relative effect of long and short boilers.

(99.) We have so far supposed that a certain fixed quantity of fuel was used with a certain fixed volume of air, but with a powerful chimney and a regulated draught, the velocity of the current of air might be so adjusted that the volume was, say, reduced to one half or doubled, &c., &c. Say we admit a double volume, producing in our case a double velocity, if a double quantity of coals were burnt in the *same time*, we should have the common case of a fire forced beyond its proper intensity; if, on the other hand, the quantity of fuel was not increased, then twice the necessary quantity of air would be used. In both cases a loss of useful effect would ensue, the amount of which we will proceed to investigate.

(100.) "*Effect of Forced Firing.*"—Taking the first case of a forced fire, and referring again to Fig. 8, we shall observe that with double velocity, the air which in a given time moved half the length of the boiler from A to B, will in our new case have moved from A to C; and as the amount of heat lost by it is *a question of time*, it follows that instead of being reduced to 552°, as at C, it will be reduced to 778° only, as at B, and will pass off into the chimney at this last temperature, bearing off with it $778° - 552° = 226°$ more heat than in the former case; so that by forced firing we lose for each pound of coal $226 \times 22 \cdot 83 \times \cdot 238 = 1228$ units of heat; and as with ordinary firing we obtained 8190 units (Table 36), we now obtain only $8190 - 1228 = 6962$ units, showing a loss of nearly 15 per cent. by forcing the fire to the extent of a double consumption of fuel.

(101.) "*Effect of too much Air.*"—When more air is used than is necessary to effect the proper combustion, a great loss occurs. We shall have as before 6175 units given out by radiation to the fire-box surrounding the fuel, and 6175 units to be carried off by the air; but, instead of 22·83 lbs. of air per lb. of coal, we shall now have 45·66 lbs. of air, its temperature will therefore be raised $\frac{6175}{45 \cdot 66 \times \cdot 238} = 569°$, and the atmospheric temperature being 62°, it will leave the fire at $569° + 62° = 631°$ instead of 1200°. Then, calculating as before with Fig. 8 and (94), we have $631° - 300° = 331$ for the first difference, and $\frac{331}{10} = 33$ for the decrease between the first and second points, the second point will therefore be $631° - 33° = 598°$, &c., &c.;

and thus we obtain the series of numbers in Fig. 10. It will be observed that as the velocity is double, the distances between point and point will be doubled, and the air will pass into the chimney at 475°, or 475° − 62° = 413° above the temperature of the atmosphere; and the amount of heat lost thereby will be 45·66 × 413 × ·238 = 4488 units, and we have—

In ashes left unburnt	650 units.
Lost by air in chimney	4488 „
Ditto by radiation, &c., in boiler-house ..	1560 „
Utilized in production of steam	6302 „
	13000 „

With the proper quantity of air, 8190 units were utilized per lb. of coal (Table 36), whereas now we have only 6302 units, showing a loss of 23 per cent. of useful effect, by the admission of double the necessary quantity of air.

(102.) This is a fruitful source of loss of fuel in very many cases: an ignorant stoker delights in a roaring fire and sharp draught, unconscious of the loss of fuel incurred; in all cases the damper should be regulated so as to produce a moderate draught, and this is especially important where there is a tall or powerful chimney.

(103.) "*Effect of too little Air.*"—Care must be taken on the other hand not to curtail the supply of air too much, as in that case also a great loss would arise from the formation of carbonic *oxyde* instead of carbonic *acid*; carbonic oxyde being formed of one equivalent of carbon and one oxygen, whereas carbonic acid is formed of carbon 1 and oxygen 2.

The experiments of Favre and Silbermann in Table 30 show that a pound of carbon burning to carbonic oxyde yields only 4453 units of heat, whereas it would yield 12906 units in burning to carbonic acid by the experiments of Dulong. The coal we have been considering (28), which is composed of ·812 carbon and ·041 hydrogen *in excess*, will give—

In carbon burning to carbonic oxyde	·812 × 4453 =	3616 units.
„ hydrogen in excess to water	·041 × 62535 =	2564 „
		6180 „

The same coal with a proper quantity of oxygen (28) gave 13044 units, we have therefore in our case only $\frac{6180}{13044} = \cdot 46$, or 46 per cent. of the available heat which the fuel could supply.

(104.) We have here taken an extreme case, the quantity of air being only half the amount absolutely necessary for proper combustion, and by (60) ¼th of the amount usually consumed in well-regulated furnaces. Where the air is curtailed, but to a less degree, part of the carbon will be tranformed into carbonic acid, and part into carbonic oxyde, and the result will be intermediate between the extremes we have given. Say we have 1½ oxygen to 1 carbon; in that case they will still combine only in the proper proportions to form one or other of the two products, that is to say, 1 to 1, or 1 to 2, but they may arrange themselves thus: we have 1½ oxygen which will divide itself into two unequal portions, 1 oxygen combining with ½ carbon, forming 1½ carbonic acid, and ½ oxygen combining with ½ carbon forming 1 carbonic oxyde, and in that case the coals we have considered would give—

In carbon burning to carbonic acid	·406 × 12906 =	5240	units.
„ „ „ oxyde	·406 × 4453 =	1808	„
„ hydrogen „ water	·041 × 62535 =	2564	„
		9612	„

In this case we should have therefore $\frac{9612}{13044} = \cdot 73$, or 73 per cent. of the heat due to the best condition with the proper quantity of air.

(105.) This will explain the anomalous fact that where there is a bad draught, not only is there difficulty in keeping up the steam, &c., but that there is a great consumption of fuel for the work done; it might be expected that with a slow dull fire few coals would be burnt, or if by dint of forcing, fuel was largely consumed, it must yield the heat due to it, but it will be evident that it is possible for there to be a dull fire, a large consumption of fuel, and little useful result at one and the same time, and all this arising from insufficient draught.

(106.) It will also be seen that in every case the proper regulation of the damper is a matter of extreme importance, and that nice adjustment is necessary to produce the best effect, too much or too little air causing a great loss of fuel; an intelligent stoker, without any knowledge of the theory, finds by experience the height of damper with which he can do the work with least fuel; if the work varies he watches and adjusts the damper accordingly, and such a man should have more consideration and better wages than he usually receives.

CHAPTER III.

ON STEAM-BOILERS.

HAVING in the preceding chapter investigated the phenomena of combustion, &c., we may apply the results to steam-boilers in practice, checking and, if needs be, modifying our deductions by the dictates of experience. We found in (98) that in all cases the most economical size of boiler was a *medium* one, and that a departure therefrom in either direction was followed by a loss of effect, an excessively long and a very short boiler giving less duty for the fuel used than a medium-sized one, properly proportioned to the work to be done.

(108.) When heated air is in contact with a surface much colder than itself, the amount of heat given out is not only a question of time, but also of *position*, of the receiving surface. Let A, Fig. 16, be a square vessel full of cold water, and let heated air pass along the four flues B C D E; the four surfaces F G H J will absorb very different quantities of heat, although they are all of the same area, &c. The surface F will receive the most, for two reasons: the hottest of the heated air will occupy the upper portion of the flue, in immediate contact with the bottom of the boiler; and the water when heated becomes lighter there, and immediately ascends, and is replaced by colder water; and so the heat received is rapidly distributed through the mass of the water. But the surface H is in the worst possible position, for if the water in contact with it is heated, it becomes lighter and remains persistently in contact with the surface, refusing to communicate heat downwards; thus it has been found by experiment that alcohol or turpentine, &c., sufficient to boil a given mass of water, may be ignited and burnt when floating on the surface of the water without increasing its temperature 1°. We may therefore admit that the surface H is useless, receiving no heat whatever. The surfaces G and J are in an intermediate position, and we may assume that they absorb a quantity of heat, a mean between F and H, or $\frac{0+1}{2} = \frac{1}{2}$; the whole boiler therefore receives only half the amount of heat that it would have received if its whole surface had been as effective as F. We have thus to consider, not only the *real* surface of a boiler, but the *effective* surface, in estimating the result to be obtained.

(109.) In Fig. 17 we have an octagonal boiler, and by the same principles we find that if the surface F absorbs an amount of heat represented by 1, B will receive 0, and C D E the respective quantities $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$, as in the figure.

(110.) Let Fig. 18 be a cylindrical boiler filled with water for illustration, as before, we have the effective surface represented by a series of numbers, as in the figure from 0 at A to 1 at B, &c.; and in all the three forms it will be seen that the effective surface is *half* the real surface where the whole is exposed to the heated air.

(111.) In an internal flue, as Fig. 19, the same series of numbers occur, but in inverted order, the most effective surface being at the top, decreasing to 0 at the bottom as in the figure.

(112.) With cylindrical boilers, heated outside, the lower half of the cylinder only is usually made available as in Fig. 19, in which case the effective surface is (Fig. 18) $\frac{\frac{1}{2}+1}{2} = \frac{3}{4}$ of the real surface exposed to the heated air, but where the whole surface is exposed, as in flues heated internally, the effective surface is half the real, as we have seen (110).

(113.) In determining the area necessary for doing a given amount of work, we must be guided by practical experience. There is great latitude here, for we have seen by (100) that a boiler, nominally of a certain power, may be forced up to double or more, with a certain loss of economy; and also that a slight increase of efficiency may be obtained by a great increase in size, up to a certain point. Besides, the area necessary per horse-power, &c., varies somewhat with the size of the boiler; a small boiler is not so effective in proportion to the area as a large one (see 98), the loss by radiation, &c., being much greater in proportion to the power.

It is found by experience that a 4-horse boiler requires about 18 *effective* square feet of surface per horse-power; 10-horse, about 14 square feet; 20-horse, about 12; and 50-horse, about 11. The best rule we can give is an empirical one, $H + (\sqrt{H} \times 2 \cdot 5) \times 8 = A$, in which H = the horse-power, and A = the effective area of the boiler in square feet. Table 38 is calculated by this rule.

(114.) The term horse-power is so indefinite that it is preferable to estimate the power of a boiler by the cubic feet of water evaporated to steam per hour; but fortunately, a common high-pressure engine of fair size, working without expansion,

TABLE 38.—Of the AREA of SURFACE for STEAM BOILERS and their FIRE GRATES.

Nominal Horse-power of Boiler.	"Effective" area of surface of Boiler in Square feet.		Area of Fire Grate in Square feet.	
	Total.	Per Horse-power.	Total.	Per Horse-power.
3	59	19·60	3·28	1·09
5	85	17·00	4·72	·94
10	143	14·32	7·94	·79
15	198	13·20	11·0	·74
20	249	12·45	13·8	·69
25	300	12·00	16·7	·67
30	350	11·67	19·4	·65
40	446	11·16	24·8	·62
45	494	10·98	27·4	·61
50	542	10·83	30·1	·60
60	635	10·58	35·3	·59

except a small amount obtained by lap of the slide, consumes about a cubic foot of water evaporated to steam per hour. Thus an engine of 12-horse power—say with 12¼-inch cylinder, 24-inches stroke, cutting off steam at 17 inches by lap of slide, with 40 revolutions per minute and 40 lbs. steam—would have 118 inches area of cylinder, and by Table 11 the volume of steam 476 for water 1, and would consume $\frac{118 \times 17 \times 2 \times 40 \times 60}{1728 \times 476} =$ 11·98 cubic feet of water per hour, or very nearly a cubic foot per horse-power.

(115.) The term horse-power is convenient, and we shall use it therefore with this limitation. We can qualify the result, for application to other kinds of engines, afterwards. By (48) it will be seen that we have assumed 60° for the temperature of the feed water, therefore a cubic foot of water at 60° evaporated to steam is equal to 1-horse power.

Table 31 gives, in col. 6, the number of pounds of different kinds of fuel per cubic foot of water or per horse-power.

(116.) When a boiler is applied for working a steam-engine, its power is usually estimated by that of the engine which it supplies, and as different kinds of engines require different quantities of steam to do the same work, it is necessary to consider that fact in fixing the size of the boilers. If we know the particulars of the engine, we can calculate approximately the amount of water it will require as in (114). Say we have a 10-horse high-pressure expansive engine, with a 12-inch cylinder, 2-feet stroke, 60 revolutions per minute, and by a double slide, cutting off its steam at $\frac{1}{3}$rd. The area of the cylinder 1 foot diameter is ·7854 of a square foot, and hence we have $\frac{\cdot 7854 \times 4 \times 60}{3} = 63$ cubic feet of steam per minute, or 3780 cubic feet per hour. The bulk of steam at say 45 lbs. per square inch is, per Table 11, 439 times the bulk of water, we have therefore $\frac{3780}{439} = 8 \cdot 7$ cubic feet of water evaporated per hour, and the size of the boiler may be calculated for that quantity, instead of 10 horses, the power of the engine.

Calculating in this way, we find by extensive observation on engines of various powers, that a common high-pressure engine, non-condensing and non-expansive, except by the slide lap, requires 1 cubic foot of water per horse-power per hour, a similar engine cutting off at $\frac{1}{3}$rd, about ·87 of a cubic foot; and a double-cylinder condensing-engine about the same. The safest course, however, is to calculate in each particular case by the capacity of the cylinder, &c., &c., and it is advisable in all cases to allow ample boiler power, see (100).

(117.) As an illustration of the application of the rule in (113), &c., say we take the case of the 20-horse boiler, Fig. 39. The tube has an effective area of $3 \cdot 14 \times 3 \times \cdot 5 \times 21 = 98$ square feet, and the body having the lower half only exposed to the heated air gives $\frac{3 \cdot 14 \times 5 \cdot 5}{2} \times \cdot 75 \times 21 = 136$ square feet effective area; so that we have a total of $98 + 136 = 234$ square feet, or $\frac{234}{20} = 11 \cdot 7$ square feet per horse-power. Table 39 gives the proportions of Cornish boilers from 3 to 55 horse power, and Figs. 36 to 43 other particulars; the thickness of plates is in accordance with the best London practice, being rather stronger than the theory indicates. For convenience, we have given with each particular boiler 3 or 4 different lengths and the corresponding horse-power.

(118.) Cornish boilers have become so common as almost to supersede all others, but the same rules apply to the many forms of tubular and other boilers occasionally used, and do not require special illustration, but it will be observed that a tube

TABLE 39.—Of the SIZES, &c., of CORNISH BOILERS, and the Thickness of Plate for 50 lbs. Steam.

Cubic feet of Water to Steam per Hour, and Horse-power in High-pressure Non-expansive Engines.	Body of Boiler.			Flue or Fire-tube.					Length of Boiler.	Effective Area of Boiler.		Fire-grate.				Diameter of Safety-valves.	
	Diameter inside.	Thickness of Plate.	Bursting Pressure in lbs. per Sq. inch.	Number of Flues.	Diameter.	Thickness of Plate.	No. of Rings.	Collapsing Pressure in lbs. per Square in.		Total.	Per Horse-power.	Length.	Width.	Area.	Area per Horse-power.	Steam 45 lbs.	Steam 7 lbs.
	ft. in.				ft. in.				ft. in	Sq. ft.	Sq ft.	ft. in.	ft. in.	feet.	feet	inches.	inches.
3	3 6	$\frac{5}{16}$	446	One	2 0	$\frac{5}{16}$	0	329	8 0	58·12	19·4	1 9	2 0	3·5	1·17	$1\frac{1}{2}$	3 one
4	..	..	..	..	..	..	..	263	10 0	72·4	18·1	2 3	..	4·5	1·12	2	$3\frac{1}{2}$,,
5	..	..	..	..	..	..	..	220	12 0	86·8	17·3	2 6	..	5·0	1·0	2	$3\frac{1}{2}$,,
5	4 0	$\frac{3}{8}$	468	One	2 3	$\frac{3}{8}$	0	349	10 0	82·5	16·5	2 3	2 3	5·0	1·0	2	$3\frac{1}{2}$,,
6	..	..	..	..	..	..	..	290	12 0	99·	16·5	2 6	..	5·62	·935	2	4 ,,
8	..	..	..	..	..	..	..	249	14 0	115	14·4	3 0	..	6·75	·844	$2\frac{1}{2}$	4 ,,
8	4 6	$\frac{7}{16}$	486	One	2 6	$\frac{3}{8}$	0	262	12 0	110	13·8	2 6	2 6	6·25	·78	$2\frac{1}{2}$	4 ,,
10	..	..	..	..	..	$\frac{7}{16}$	..	293	15 0	138	13·8	3 0	..	7·5	·75	$2\frac{1}{2}$	$4\frac{1}{2}$,,
12	..	..	..	..	..	..	..	258	17 0	157	13·1	3 6	..	8·75	·73	$2\frac{1}{2}$	5 ,,
10	5 0	$\frac{7}{16}$	437	One	2 9	$\frac{7}{16}$	0	307	13 0	133	13·3	2 9	2 9	7·56	·75	$2\frac{1}{2}$	$4\frac{1}{2}$,,
12	..	..	..	..	..	..	..	246	16 0	163	13·6	3 3	..	9·0	·75	$2\frac{1}{2}$	5 ,,
14	..	..	..	..	..	$\frac{3}{8}$	One	317	18 0	184	13·1	3 9	..	10·3	·73	3	$5\frac{1}{2}$,,
16	..	..	..	..	..	..	..	285	20 0	204	12·8	4 3	..	11·7	·73	3	$5\frac{1}{2}$,,
15	5 6	$\frac{7}{16}$	397	One	3 0	$\frac{3}{8}$	One	316	16 0	179	11·9	3 6	3 0	10·5	·7	3	$5\frac{1}{2}$,,
18	..	..	..	..	..	$\frac{7}{16}$	..	386	19 0	212	11·8	4 0	..	12·0	·67	3	$5\frac{1}{2}$,,
20	..	..	..	..	..	..	..	334	21 0	234	11·7	4 6	..	13·5	·67	3	6 ,,
22	..	..	..	..	..	..	..	306	23 0	256	11·6	5 0	..	15·0	·68	3	$4\frac{1}{2}$ two
22	6 0	$\frac{1}{2}$	416	Two	2 3	$\frac{3}{8}$	One	342	18 0	255	11·6	3 3	4 6	14·6	·66	3	$4\frac{1}{2}$,,
24	..	..	..	..	..	..	..	337	20 0	282	11·7	3 6	..	15·75	·66	$3\frac{1}{2}$	$4\frac{1}{2}$,,
27	..	..	..	..	..	..	..	307	22 0	310	11·5	4 0	..	18·0	·66	$3\frac{1}{2}$	5 ,,
30	..	..	..	..	..	..	..	281	24 0	338	11·3	4 3	..	19·1	·64	$3\frac{1}{2}$	5 ,,
30	6 6	$\frac{1}{2}$	335	Two	2 6	$\frac{3}{8}$	One	285	22 0	341	11·4	3 9	5 0	18·75	·62	$3\frac{1}{2}$	$5\frac{1}{2}$,,
35	..	..	..	..	..	..	Two	375	25 0	387	11·07	4 6	..	22·5	·64	4	$5\frac{1}{2}$,,
40	..	..	..	..	..	..	Two	336	28 0	434	10·85	5 0	..	25·0	·62	4	6 ,,
45	7 6	$\frac{9}{16}$	401	Two	3 0	$\frac{7}{16}$	Two	408	27 0	488	10·8	4 6	6 0	27·0	·6	$4\frac{1}{2}$	6 ,,
50	..	..	..	..	..	..	..	366	30 0	542	10·8	5 0	6 0	30·0	·6	$4\frac{1}{2}$	$6\frac{1}{2}$,,
55	..	..	..	..	..	..	..	333	33 0	596	10·8	5 6	6 0	33·0	·6	5	$6\frac{1}{2}$,,

with heated air inside receiving heat all over is more effective than the cylindrical body of a boiler heated up to its centre only as in Fig. 19; taking for example equal diameters, say 1 foot, a flue heated internally exposes an *actual* surface of 3·14 square feet per foot in length, or $\frac{3 \cdot 14}{2} = 1 \cdot 57$ feet *effective;* but the same tube heated outside up to the centre gives an actual area of $\frac{3 \cdot 14}{2} = 1 \cdot 57$ feet, and an effective area of $1 \cdot 57 \times \cdot 75 = 1 \cdot 18$ square feet only. A tubular boiler is therefore more effective than an ordinary one with the same actual surface.

(119.) "*Furnaces.*"—The consumption of fuel per square foot of grate may be varied very considerably without any sensible effect on the economy; a *very* large grate and thin fire is very objectionable, for without great care, parts of the grate are liable to become uncovered, whereby a large volume of air passes through the furnace unconsumed, resulting in a great loss of useful effect (101).

(120.) It has been found by experience that in ordinary cases 12 or 14 pounds of coal per square foot of grate per hour is a good average quantity. By Table 31, we find that with Newcastle coals we require 8·46 lbs. of coal per horse-power, for a 20-horse boiler we should therefore require $8 \cdot 46 \times 20 = 169$ lbs. of coal per hour, and taking a mean consumption of 13 lbs. per square foot of grate, we have $\frac{169}{13} = 13$ square feet of grate, or ·65 square feet per horse-power. But this will not be the same for all powers of boilers for the reasons given in (113); in fact, the area of the fire-grate should be proportional to the area of the boiler, say $\frac{1}{18}$th of the *effective* area, and thus we have calculated col. 4 in Table 38. The 20-horse boiler, having by (117) an area of 234 feet, should have about $\frac{234}{18} = 13$ square feet of fire-grate.

(121.) "*Fire-Bars.*"—Fire-bars should be short, thin, and deep; the length in most cases should not exceed 3 feet, a long fire-bar is apt to be distorted by the heat, and give trouble, and this will not be obviated by increasing the thickness. A thin bar will stand better than a thick one, despite its apparent weakness, but the fact is, that a fire-bar is cooled by the passage of cold air on both sides of it, and thus a thin bar is cooled more effectively than a thick one. Fig. 12 gives good general proportions for fire-bars; the dimensions apply to all lengths except the depth at the centre, which is given by Table 40,

TABLE 40.—Of the PROPORTIONS of FIRE-BARS for STEAM-BOILERS, &c., &c.

Length.		Depth.	Weight.	Weight per Square foot.	Length.		Depth.	Weight.	Weight per Square foot.
ft.	in.	inches	lbs.	lbs.	ft.	in.	inches.	lbs.	lbs.
1	0	2½	6·4	64	3	3	4¾	29	89
1	3	2¾	8·1	64	3	6	5	32	91
1	6	3	9·5	64	3	9	5¼	35	93
1	9	3¼	11·4	65	4	0	5½	39	98
2	0	3½	*13	65	4	3	5¾	42	100
2	3	3¾	*15½	69	4	6	6	47	104
2	6	4	*18	72	4	9	6¼	52	109
2	9	4¼	*20¾	70	5	0	6½	*56¾	113·5
3	0	4½	*23¼	79					

which is calculated by the rule $L + 1 \cdot 5 = d$, in which L = the length of bar in feet, and d the depth at the centre in inches. The fourth column shows that short bars weigh less per *square* foot than long ones.

"*Dead-plate, &c.*"—The object of the dead-plate is principally to keep the fire away from the furnace-front, and prevent it becoming unduly heated. Its width may vary with the size of the boiler, as is shown by Figs. 36 to 43, being 6 inches wide in small, 9 inches in medium, and 12 inches in large boilers. For the same purpose, a screen-plate as at D in Fig. 8 is a useful addition to the furnace door. The figures give the longitudinal section of furnaces. The bearing-bars should not be firmly fixed to the boiler, as at A, Fig. 20, as is frequently done, for in that case the expansion by heat is very apt to work the screws loose, and cause leakage; but should rest loosely in a pocket formed of angle-iron, as at B. The back-bridge may be of thick cast-iron, or of fire-brick, as in the figures.

(122.) "*Steam-chest.*"—The primary object of a steam-chest is to form a steam reservoir, where the steam may be quietly separated from the water, which otherwise is very apt to *prime* over with it (46.) In single-flued Cornish boilers, the steam-space is very small, and a steam-chest is essential; but in the double-flued boilers, as in Fig. 36, &c., there is abundant steam-room without a steam-chest: nevertheless it is advisable to adopt it in all cases; for with a little management, the man-hole and the steam-valves, &c., may all be fixed upon it, as in Fig. 21, and in that case no openings whatever are required in the body of the boiler. The whole may be floored over with York paving, and great neatness obtained. This cannot, however, be done very well with less diameter than 2 ft. 6 in. The top of the chest will require for high-pressure steam a bar of T iron across to strengthen it, a second man-hole must be made in the body of the boiler, immediately beneath the one in the top of the chest; but in no case must the whole of the metal in the body covered by the chest be cut out, as the boiler would thereby be seriously weakened.

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* Actual weights, obtained by weighing the castings.

(123.) "*Safety-valves.*"—The velocity with which steam issues from a boiler, &c., is the same as that of a body falling by gravity from the height of a homogeneous column of steam, having throughout the same density as at the orifice. Say we take the case of steam 40 lbs. per square inch above the atmosphere. By Table 41, one pound pressure is equal to 2·3 feet of water, therefore the pressure in our case is $2 \cdot 3 \times 40 = 92$ feet of water. By Table 11, steam at 40 lbs. is 476 times the bulk of water; the height of the column of steam is therefore $92 \times 476 = 43792$ feet, and the velocity due to that height by the rule for falling bodies ($\sqrt{H} \times 8$) is, in our case, $\sqrt{43792} \times 8 = 1672$ feet per second; but when the orifice is made in a thin plate, the issuing jet of steam suffers a contraction, so that its area is reduced, according to the experiments of Daubuisson, to ·65, the actual area being 1·0; the discharge by an aperture 1 inch square, per hour, is therefore $\frac{1672 \times 3600 \times \cdot 65}{144} = 27170$ cubic feet of 40 lbs. steam,—equal to $\frac{27170}{476} = 57 \cdot 08$ cubic feet of water, or 57-horse power according to (115).

(124.) An experiment was made with a double-flued Cornish boiler 24 ft 3 in. long, 6 ft. 6 in. diameter, with two 2 ft. 6 in. flues, and 30 ft. of fire grate (about 35-horse power, by our Table 39). The steam was discharged by an aperture 1 inch square in a thin plate, and by *forced* firing 57·5 cubic feet of water were evaporated in an hour—the pressure of steam varying from 35 to 43 lbs., agreeing remarkably with the preceding calculation. A 3¾ in. safety-valve on the same boiler, loaded to 30 lbs., discharged the same quantity of steam at 46 lbs. pressure. The valve whose angle was 45° was raised vertically ·122 inch,

TABLE 41.—Of EQUIVALENT PRESSURES in POUNDS per SQUARE INCH, FEET of WATER, and INCHES of MERCURY, at a Temperature of 62° Fahr.

Pounds per Square inch.	Feet of Water.	Inches of Mercury.	Pounds per Square Inch.	Feet of Water.	Inches of Mercury.
1·	2·311	2·046	2·5962	6·	5·31198
2·	4·622	4·092	3·0289	7·	6·19731
3·	6·933	6·138	3·4616	8·	7·08264
4·	9·244	8·184	3·8942	9·	7·96797
5·	11·555	10·230	·48875	1·12952	1·
6·	13·866	12·276	·97750	2·25904	2·
7·	16·177	14·322	1·46625	3·38856	3·
8·	18·488	16·368	1·95500	4·51808	4·
9·	20·800	18·414	2·44375	5·64760	5·
·4327	1·	·88533	2·93250	6·77712	6·
·8654	2·	1·77066	3·42125	7·90664	7·
1·2981	3·	2·65599	3·91000	9·03616	8·
1·7308	4·	3·54132	4·39875	10·16568	9·
2·1635	5·	4·42665			

EXAMPLE.—Required the Pressure per Square Inch, and Equivalent Column of Mercury for a Head of 247 feet of Water.

Feet of Water.		Pounds per Square Inch.		Inches of Mercury.
200	=	86·54	or	177·066
40	=	17·308	„	35·413
7	=	3·029	„	6·197
247	=	106·877	„	218·676

as in Fig. 22; the width of the annular discharging-orifice was therefore $\frac{\cdot 122}{1\cdot 414} = \cdot 0863$: and the circumference of 3¾ being 11·78, we have ·0863 × 11·78 = 1·016 square inches as the area, or very nearly the same as before. It will be observed that although the valve was loaded to 30 lbs., and would doubtless begin to blow off at that pressure, yet the pressure increased to 46 lbs. before the valve was able to carry off the steam, giving an increase of more than 50 per cent. This great increase arises from the fact that the valve was too small for the quantity of steam. It was large enough for the boiler when working at its nominal power of 35 horses, but not for 57·5-horse power to which it was forced; for, allowing 11 square feet per horse-power as in (113), a boiler for that quantity should have about

640 square feet area, requiring, as per Table 42, a 5-inch safety-valve. Even with that size, the pressure would increase considerably (perhaps 15 or 20 per cent.) above that to which the valve was loaded. Nor can this be obviated without using valves of enormous and impracticable size.

(125.) It will be seen from this that there is no precise standard for the size of a safety-valve: all that can be done is to fix upon a size that will not suffer the pressure to rise to a dangerous extent. The area of a safety-valve should be proportional to the area of fire-surface in the boiler, and should be determined by that rather than by the horse-power of the boiler. We may admit as the result of experience that a 20-horse boiler (as in Table 39) with 45 lbs. steam requires one valve 3 inches diameter. Now that boiler has an area of 234 square feet, or $\frac{234}{3^2}$ $= 26$ square feet per circular inch of valve, or very nearly 3 square yards (27 feet.); and for this pressure we may take the rule $A = d^2 \times 26$, in which $d =$ the diameter of the valve in inches, and $A =$ the *effective* area of the boiler in square feet. The fourth column in Table 42 has been calculated by this rule.

(126.) We can calculate from this the areas with other pressures, say with the same boiler, we take 7 lbs. steam. We find by Table 11 that we have to discharge a larger volume of steam than with 45 lbs. pressure, in the ratio of 1138 to 439, so that the 3-inch valve, which was equal to 243 square feet of boiler surface with 45 lbs. steam, would now be equal to only $\frac{243 \times 439}{1138} = 93 \cdot 7$ square feet, if the velocity of discharge was the same, which it is not, for (123) with 45 lbs. steam the velocity is $\sqrt{45 \times 2 \cdot 3 \times 439 \times 8} = 1704$ feet per second, but with 7 lbs. steam, $\sqrt{7 \times 2 \cdot 3 \times 1138 \times 8} = 1090$ feet per second, and the area of boiler surface for a 3-inch valve is thus reduced to $\frac{93 \cdot 7 \times 1090}{1704} = 60$ square feet, or about one-fourth of the area with 45 lbs. steam. The different columns in Table 42 have been calculated in this way. Table 39 also gives the proper diameter of safety-valve for different sizes of boilers.

(127.) The forms of safety-valves are arbitrary; a convenient and inexpensive form is given by Fig. 25. They are sometimes made close-topped, with a waste-pipe to the chimney, &c., but this is objectionable, because the valve may be leaky, or by careless firing the pressure may frequently be allowed to get up too

TABLE 42.—Of the SIZES of SAFETY VALVES for STEAM BOILERS.

Diameter of Valve.	Pressure of Steam in Pounds per Square Inch above the Atmosphere.				
	7	25	45	65	100
	"*Effective*" area of Boiler surface in Square Feet.				
In.					
1	7	17	27	38	54
1½	15	39	61	84	123
2	27	67	108	149	218
2½	42	105	170	235	343
3	61	151	243	335	491
3½	83	205	331	457	668
4	108	268	432	596	873
4½	137	329	547	755	1105
5	169	418	675	932	1363
5½	203	505	814	1123	1644
6	243	603	972	1341	1963
	Velocity of discharge in Feet per Second.				
	1090	1560	1704	1784	1872

NOTE.—The *effective* area meant in this Table is explained in (108–113); see also Table 38 for the connection between the effective area and the horse-powers of boilers, &c.

high, and blow off, without attracting attention. The open-topped form is preferable, and with ordinary care in firing should seldom be found blowing off. In adjusting the weight, allowance should be made for the weight of the lever and valve; this can be done by the application of a Salter's balance at B. Say we had a 3-inch valve for 45 lbs. steam, and that the *effective* weight at B was 12 lbs., and we had to determine the weight at C, the distances A B and A C being 3¼ and 19·5 inches respectively, or 1 to 6. The area of 3 inches being 7·06, we have $\frac{(7\cdot06 \times 45) - (12) \times 3\cdot75}{19\cdot5}$ or $\frac{(7\cdot06 \times 45) - 12}{6} =$ 51 lbs. at C. The seat of the valve should not be wide, as in Fig 22, but very narrow, as in Fig. 26; with a wide seat, there is considerable uncertainty in estimating the actual or effective diameter, which is usually intermediate between the large and small diameters of the cone; with a narrow seat this is avoided, and if well executed it is more easy to grind true and keep steam-tight.

(128.) "*Dampers.*"—The area of a damper depends on that of the chimney, and where there is only one boiler it may have the same area as the chimney, if that is properly proportioned to the power of the boiler. Thus for a chimney, as per Table 55, 40 feet high, 12 inches square, we have $\frac{144}{8\cdot1} = 17\cdot7$ inches per horse-power, but for a chimney 150 feet high, and 2 feet 6 inches square, $\frac{900}{108} = 8\cdot4$ inches per horse-power only. As an approximate rule, we may give $\frac{110}{\sqrt{H}} = A$, in which H = the height of the chimney in feet, and A = the area of the damper in square inches per horse-power; thus for a chimney 100 feet high we have $\frac{110}{10} = 11$ inches per horse-power. The form of damper is arbitrary, and must often be varied to suit the form of the flue, but for ordinary cases we may adopt standard sizes, a convenient proportion being 3 to 1, and thus we have the sizes and powers given in Table 43. The powers of other sizes may be easily

TABLE 43.—Of the SIZES of DAMPERS to STEAM BOILERS, with different Heights of Chimney.

Size of the Damper in Inches.	Height of Chimney in Feet.					
	40	60	80	100	120	150
	Square Inches of Damper per Horse-power.					
	17·4	14·2	12·4	11·0	10·0	9·0
	Horse-power of the Boiler.					
6 × 18	6·2	7·6	8·7	9·9	10·8	12
7 × 21	8·5	10	12	13	15	16
8 × 24	11	13	16	18	19	22
9 × 27	14	17	20	22	24	27
10 × 30	17	21	25	28	30	34
12 × 36	25	31	35	40	43	48
14 × 42	34	41	47	53	59	65
16 × 48	44	54	62	70	77	85

calculated by the numbers in the fourth line of the table; thus, say we required the size for a large damper to a set of boilers 300-horse power for a chimney 100 feet high; the table gives 11 inches per horse-power, and we have $300 \times 11 = 3300$ square inches for the area required, and if the height was fixed at 6 feet, or 72 inches, the width must be $\frac{3300}{72} = 46$ inches, &c., &c.

(129.) "*Area and Arrangement of Flues.*"—The volume of heated air which has to pass along a boiler flue is proportional to the horse-power, and as in order to give out a given amount of heat, or to be cooled to a fixed temperature, it must be in contact with the boiler a certain *time*, it follows that the velocity of the current should be proportional to the length of the boiler, so that a particle of air traverses it in the same time, whether the boiler be long or short, and departs into the chimney cooled down to the same temperature in all cases. Let us take three boilers, say 10, 20, and 30 feet long, and 4, 16, and 50-horse power, as per Table 39; the velocities should therefore be in the ratios 1, 2, 3, and the volumes of air in the ratios 4, 16, 50, or 1, 4, 12 nearly; the areas must therefore be in the ratio of $\frac{\text{Volume}}{\text{Velocity}}$ or $\frac{1}{1} = 1$, $\frac{4}{2} = 2$, and $\frac{12}{3} = 4$, so that the 50-horse boiler requires a flue only four times the area of a 4-horse one with the length of boilers we have taken. If the volume of air is proportional to the horse-power, and the velocity proportional to the length, the area of flue would be in the ratio of $\frac{\text{Horse-power}}{\text{Length}}$.

(130.) But we have seen (113) that the volume of air is not exactly proportional to the horse-power, small boilers consuming more fuel than large ones, and requiring more air. The area of flue must therefore be made proportional to the effective area of the boiler, rather than its horse-power, and the rule becomes $\frac{A}{L} \times 47 = a$, in which A = the *effective* area of the boiler in square feet, as in (108), &c., L = the length of boiler in feet, and a = the area of flue in square inches. Thus in our cases the 4-horse boiler requires $\frac{72 \cdot 4}{10} \times 47 = 348$ square inches, or 87 inches per horse-power; the 16-horse $\frac{204}{20} \times 47 = 480$ inches or 30 inches per horse-power; and the 50-horse $\frac{542}{30} \times 47 = 851$ inches, or 17 inches per horse-power.

(131.) There are three principal ways in which the flues of a Cornish boiler may be arranged. Fig. 27 shows the best plan: the fire proceeding along the tube to the back of the boiler descends and returns beneath the body to the front, where it splits and passes on both sides to the chimney. This mode of setting is preferable to any other, because the *bottom* of the boiler is more effectively heated, and thereby a better circulation of the water is effected; it requires, however, rather more length in the house than other modes. Fig. 28 is another arrangement, in which the fire splits at the back, returning on both sides to the front, where it descends, and proceeds by one flue to the chimney. Fig. 29 shows a "wheel" draught, and is the most common form of any. The fire returns to the front on one side, passes under the boiler by an opening in the dividing-wall, and passes to the chimney on the other side. The flues which receive the first part of the heat should be lined with fire-brick; the boiler should be fixed at such a level that the fire-bars are about 2 feet 6 inches above the stoke-hole floor.

(132.) "*The Strength of Steam Boilers.*"—For almost all our knowledge of the strength of steam boilers we are indebted to the venerable Mr. Fairbairn. From his experiments the mean strength of boiler-plate is 23 tons, or 51520 lbs. per square inch, but the strength with ordinary single-riveted joints is 66 per cent. of the strength of a solid plate, or 34000 lbs. per square inch, and this has commonly been taken as a standard in calculating the strength of boilers; but Staffordshire plates, which are extensively used by the best makers, bear only 20 tons per square inch, or $20 \times \cdot 66 = 13 \cdot 2$ tons, or say 30,000 lbs. per square inch with single-riveted joints. Hence $\frac{30000 \times 2t}{d} =$ P, in which t = thickness of plate in inches, d = the internal diameter of the boiler in inches, and P the bursting-pressure in pounds per square inch.

Table 44 has been calculated by this rule for the thickness in the first column. The *working-pressure* may be taken at one-sixth of the bursting-pressure.

TABLE 44.—Of the STRENGTH of CYLINDRICAL BOILERS, made of Staffordshire Plates and Single-riveted Joints to resist Internal or Bursting Pressure.

Thickness of Plate.		DIAMETER OF BOILER (INSIDE) IN FEET AND INCHES.													
		ft. in. 1 6	ft. in. 2 0	ft. in. 2 6	ft. in. 3 0	ft. in 3 6	ft. in. 4 0	ft. in. 4 6	ft. in. 5 0	ft. in. 5 6	ft. in 6 0	ft. in. 6 6	ft. in. 7 0	ft. in. 7 6	ft. in. 8 0
Theoretical.	Practical.	BURSTING-PRESSURE IN POUNDS PER SQUARE INCH.													
$\frac{1}{16}$	$\frac{3}{16}$	208	156	125	104	89	78	69	62	57	52	48	45	42	39
$\frac{1}{8}$	$\frac{1}{4}$	416	312	250	208	178	156	139	125	114	104	96	89	83	78
$\frac{3}{16}$	$\frac{5}{16}$	624	468	375	312	267	234	208	187	170	156	144	134	125	117
$\frac{1}{4}$	$\frac{3}{8}$	852	624	500	416	357	312	277	250	227	208	192	178	167	156
$\frac{5}{16}$	$\frac{7}{16}$	1040	780	625	520	446	390	347	312	284	260	240	223	208	195
$\frac{3}{8}$	$\frac{1}{2}$	1248	936	750	624	535	468	416	375	341	312	288	267	250	234
$\frac{7}{16}$	$\frac{9}{16}$	1456	1092	875	728	624	546	486	437	397	364	337	312	292	273
$\frac{1}{2}$	$\frac{5}{8}$	1664	1248	1000	832	714	624	555	500	454	416	385	357	334	312
$\frac{9}{16}$	$\frac{11}{16}$	1872	1404	1125	936	803	702	624	562	511	468	433	401	375	351
$\frac{5}{8}$	$\frac{3}{4}$	2080	1560	1250	1040	892	780	694	625	568	520	481	446	417	390
$\frac{11}{16}$	$\frac{13}{16}$	2288	1716	1375	1144	981	858	763	687	625	572	529	491	458	429
$\frac{3}{4}$	$\frac{7}{8}$	2496	1872	1500	1248	1070	936	833	750	681	624	577	535	500	468

(133.) This rule shows, that in theory the thickness of boilers of different diameters, for the same pressure, should vary simply with the diameter; for instance, a 6-ft. boiler for high-pressure steam of 50 lbs. might be as per Table, $\frac{3}{8}$ thick, a 3-ft. one being $\frac{3}{16}$, and an 18-in. one, $\frac{3}{32}$, &c., but these strengths would in practice be considered much too light, especially for the small diameters, in fact such boilers would be apt to collapse with their own weight. For this and other reasons it is advisable to add, say $\frac{1}{8}$ of an inch, to all the thicknesses given by the Table, which is done in column 2, by which we find that for 50 lbs. safe, or 300 lbs. bursting-pressure, a 6-feet boiler should be $\frac{1}{2}$ in. thick, 4 ft. = $\frac{3}{8}$, 3 ft. = $\frac{5}{16}$, and 2 ft. = $\frac{1}{4}$ in., &c. But even with this correction the thicknesses come out too light with *very low pressures*, and in such cases the engineer must be guided rather by his judgment. Our Table gives the theoretical strength in any case, thus, with the sizes now given the bursting-strain is 416, 468, 520, and 624 lbs. per square inch respectively.

(134.) "*Strength of Boiler Tubes to resist External Pressure.*" —From the experiments of Fairbairn it appears that the strength of cylindrical tubes of boiler-plates to resist external pressure varies directly as the 2·19 power of the thickness, and inversely as the diameter and length, and he gives the following rules, in which

p = the collapsing Pressure, in pounds per square inch.
L = the Length of the boiler, in feet.
D = Diameter of ditto, in inches
t = thickness of plate, in inches

$$\text{then } p = 806300 \times \frac{t^{2\cdot19}}{\text{L}\cdot\text{D}}$$

A convenient modification of this rule is

$$p = 33\cdot61 \times \frac{(100t)^{2\cdot19}}{\text{L}\cdot\text{D}}$$

or logarithmically $1\cdot5265 + 2\cdot19\ (\log.\ 100t) - \log.\ (\text{L} \times \text{D}) = \log.\ p$.

It appears from this that $p \times \text{L} \times \text{D}$ is constant for the same thickness of plate, and its value will be given by the rule $33\cdot61 \times (100\ t)^{2\cdot19} = p \times \text{L} \times \text{D}$, or by logarithms.

$$1\cdot5265 + 2\cdot19 \times (\log.\ 100\ t) = \log.\ p \times \text{L} \times \text{D}.$$

By this rule Table 45 has been calculated, and from it we can easily calculate the strength of a tube by the simple rules of arithmetic, thus, a tube 30 in. diameter, $\frac{3}{8}$ in. thick, and 12 ft. long, will collapse with $\frac{94110}{30 \times 12} = 261$ lbs. per square inch; and again, to ascertain the proper thickness for a flue say 33 inches diameter, 20 feet long, to sustain a working-pressure of 45 lbs. per inch, or say $45 \times 6 = 270$ lbs. collapsing pressure, then $p \times \mathrm{L} \times \mathrm{D}$ is in our case $270 \times 20 \times 33 = 178200$, the nearest number to which in Table 45 is opposite $\frac{1}{2}$ inch, the thickness required.

TABLE 45.—For the STRENGTH of BOILER TUBES to RESIST EXTERNAL or COLLAPSING PRESSURE.

Thickness of Plate.	Value of $p \times \mathrm{L} \times \mathrm{D}$.	Thickness of Plate.	Value of $p \times \mathrm{L} \times \mathrm{D}$.
inch.		inch.	
$\frac{1}{16}$	1860	$\frac{9}{16}$	228700
$\frac{1}{8}$	8484	$\frac{5}{8}$	288000
$\frac{3}{16}$	20620	$\frac{11}{16}$	354900
$\frac{1}{4}$	38720	$\frac{3}{4}$	429500
$\frac{5}{16}$	63130	$\frac{13}{16}$	511700
$\frac{3}{8}$	94110	$\frac{7}{8}$	601800
$\frac{7}{16}$	131900	$\frac{15}{16}$	700100
$\frac{1}{2}$	176700	1	806300

(135.) From experiments on the large scale it would appear that *lap-jointed* tubes, such as are commonly used in practice, resist a collapsing strain more powerfully than the small experimental tubes from which the constants in the formula are derived. Fairbairn gives two experiments on large boilers, one 35 feet long with 3 feet 6 inch tubes, $\frac{3}{8}$ thick, which gave way with 97 lbs. per square inch, but by rule and Table it should have borne $\frac{94110}{42 \times 35} = 64$ lbs. only. The other experiment was on a tube 25 feet long, 3 feet 6 inches diameter, and $\frac{3}{8}$ thick, which collapsed with 127 lbs. per square inch, but whose strength by Table, &c., was $\frac{94110}{42 \times 25} = 89 \cdot 6$ lbs. only. This is only what might be expected, the double thickness at the joint acts partially like a series of rings, and increases the strength in the same way; the rules therefore give a minimum and safe strength, in fact only about 70 per cent. of the strain which ordinary boiler tubes seem to be capable of bearing.

(136.) An obvious and easy mode of increasing the strength of a boiler tube is by adding strong rings at intervals, so as (in effect) to divide it into short lengths, and Mr. Fairbairn proposes to do this by making the tube butt-jointed and using ⊥ iron for the junctions; but a butt-joint cannot easily be made tight, and it is better to make the tube lap-jointed as usual, and add T or L rings where required. Even this is objectionable from the great thickness of metal where the rings are riveted on, the metal there being in consequence unduly heated. An ingenious plan of Mr. Bramwell, C.E., of London, promises well to get over this objection:—let Fig. 30 be a cylindrical tube A B C D, compressed into the ellipsis E F G H. This change of form may be resisted in two ways, namely, by preventing E and F from giving in, or G H from giving out; if G H be prevented from bulging out, E and F will be effectually prevented from collapsing. This is effected by putting on the tube, thin deep rings of iron, say $\frac{5}{8} \times 3$ inches deep, as J in Fig. 31, such rings are free (from their small thickness) to the objection of increasing the thickness of the metal exposed to the fire, &c., &c., to which ⊥ iron rings are liable.

(137.) Elliptical tubes are exceeding weak for resisting an external pressure. Experiments were made by Mr. Fairbairn on such tubes, one 14 in. × $10\frac{1}{4}$ in. diameter, 5 feet long and ·043 thick, collapsed with $6\frac{1}{2}$ lbs. per square inch; another $20\frac{3}{4}$ in. × $15\frac{1}{2}$ in. × 5 ft. 1 in. long and ·25 thick, collapsed with $127\frac{1}{2}$ lbs. per square inch; from which it appears that in elliptical tubes the diameter to be taken in calculating the strength is $\frac{(2L)^2}{S}$, in which L = the major and S the minor axis of the ellipse; thus in Fig. 30, $L = 3$ feet and $S = 2$ feet, and the flattest part of the ellipse at E or F is part of a circle whose diameter is $\frac{(2 \times 3)^2}{2} = 9$ feet, and the collapsing pressure will be that due to a tube 9 feet diameter; this will show the excessive weakness of elliptical, or oval tubes.

(138.) The French Government and Mr. Fairbairn assume the safe working-strain on a boiler at one-sixth of the bursting or collapsing pressure, and we may admit this to be a safe rule for all ordinary cases in practice.

(139.) The flat ends of boilers require to be well stayed with longitudinal rods, or gusset stays to the body as in the Figures, these latter are every way the best and most convenient.

CHAPTER IV.

ON THE EFFLUX OF COMPRESSED AIR, GAS, AND STEAM.

(140.) "*Efflux of Compressed Air, &c.*"—When water or other liquid escapes from an orifice in the side of a vessel into the air as at A, Fig. 32, the velocity of efflux is the same as that of a body falling freely by gravity from the height S A. Similarly when a liquid escapes from one vessel into another, by a submerged orifice B, the velocity of efflux is that due to the height S T, or the *difference* of level of the liquids in the two vessels, and it is not affected by the depth T B, at which the orifice is placed.

(141.) We have here supposed that one and the same liquid was being dealt with, but if one compartment were filled with a liquid of different specific gravity to that in the other, we have a different case. Say in Fig. 33 we have a vessel with two compartments, C and D, filled to the *same level* with two fluids, whose specific gravity (for the sake of illustration) was as 3 to 1, C being the denser of the two. The conditions of pressure at E are precisely the same as would arise with one and the same fluid in both compartments, by columns having the respective heights F E and H E, that is to say, the velocity of efflux will be that of a falling body, with the height G H, and of course it is the denser fluid which escapes with that velocity.

(142.) " *Velocity into a Vacuum.*"—Applying this reasoning to elastic fluids, we are met by the difficulty that we have no real surface level to calculate from; in the case of air, for instance, the density diminishes as we leave the earth in geometric ratio, and the limit is in infinity. But for the purpose of calculation, we may find what the height of the atmosphere would be, if it had throughout the same density as it has at the surface of the earth. Assuming that the barometer was at 30 inches, we find from Table 46 the density of mercury to be 13·596, water being 1; and from Table 32 air has a density of ·001221, water being 1, the height of a homogeneous column of air equal to 30 inches of mercury is therefore $\frac{13\cdot596 \times 30}{\cdot001221 \times 12} = 27838$ feet, and although this is fictitious, we may use it for the purposes of calculation without error. We shall now find the velocity into a vacuum by the rule for falling bodies to be $\sqrt{27838} \times 8 = 1344$ feet per second.

TABLE 46.—Of the SPECIFIC GRAVITY and WEIGHT of MATERIALS, Water at 62° being 1·000.

	Specific Gravity.	Weight of a Cubic Foot in Pounds.	Weight of a Cubic Inch in Pounds.	No. of Cubic Feet in one Ton.
Mercury	13·596	847·3	·4903	2·644
Lead	11·352	707·5	·4094	3·166
Copper, sheet	8·785	547·5	·3168	4·091
Gun Metal, cast	8·670	540·3	·3127	4·145
Copper, cast	8·607	536·4	·3104	4·176
Brass, ,,	8·393	523·1	·3027	4·282
Wrought Iron	7·788	485·3	·2809	4·615
Tin, cast	7·291	454·4	·2630	4·930
Zinc, sheet	7·190	448·1	·2593	4·999
Cast Iron, British, mean	7·087	441·6	·2556	5·07
Zinc, cast	6·861	427·6	·2474	5·24
Slate	2·835	176·7	·1022	12·68
Glass	2·76	172·0	·0995	13·02
Granite, Cornish	2·662	165·9	·0960	13·50
Sandstone, Yorkshire	2·506	156·2	·0904	14·34
Brick, London Stock	1·841	114·7	·0664	19·52
Sand, River	1·546	96·35	·0558	23·25
Coal, British, mean	1·313	81·83	·0474	27·37
Water, distilled	1·000	62·321	·03606	35·95
Ice, at 32°	·93	57·96	·03354	38·65
Alcohol	·813	50·67	·02932	44·21
Oil, Olive	·9153	57·04	·03301	39·27
Oak, seasoned	·777	48·42	·02802	46·26
Elm, ,,	·588	36·65	·0212	61·13
Mahogany, Honduras, seasoned	·560	34·9	·0202	64·18
Pine, Yellow, seasoned	·483	30·1	·01742	74·41
Coke, Gas, in measure	·353	22·0	·01273	101·8
Cork	·24	14·96	·00866	149·7

(143.) The velocity of steam at atmospheric pressure into a vacuum, may be calculated in the same way: taking its density from Table 32 at ·0007613, we have $\frac{13\cdot596 \times 30}{\cdot0007613 \times 12} = 44648$ feet for the height of a column of steam equivalent to 30 inches of mercury, and the velocity into a vacuum $\sqrt{44648} \times 8 = 1690$ feet per second.

(144.) Applying these rules to air, steam, &c., of other than ordinary densities and pressures, we are conducted to the remarkable fact that the velocity into a vacuum is constant, whatever the pressure may be; for instance, air of double the

atmospheric pressure would have a double height of column, and thereby an increased velocity, if the density remained the same, but the density being of necessity double also, the height of column remains the same, and hence the velocity which is due to that height remains the same also. It follows from this, that if we filled a vessel with air compressed to any number of atmospheres and allowed it to escape into a vacuum, the velocity would be the same from first to last, although the pressure would be continuously reduced by the escape of the compressed air; but the *quantity* or weight of air which escapes would not be the same at all pressures, but would vary with the density of the air, which varies every moment with the pressure.

(145.) "*Velocity into Air.*"—This uniformity of velocity at all pressures does not hold when the discharge is made into air. Let Fig. 33 represent the discharge into rarefied air, say of one-third the ordinary density (made so by heat or otherwise), we have then a case analogous to (141), in which we found the velocity to be that due to the difference of height of the two columns. Thus in Fig. 33 we have two columns of the same height, or 27,838 feet, but the pressure exerted at the orifice E by the column G E is the same as that of a column of ordinary air of one-third the height, or $\frac{27838}{3} = 9279$ feet, and, as in (141) the velocity of efflux will be that due to the difference, or $27838 - 9279 = 18559$ feet head, namely, $\sqrt{18559} \times 8 = 1090$ feet per second.

Again, the velocity of steam of 20 lbs. pressure per square inch into the atmosphere may be calculated in the same way. By Table 41 we find that 20 lbs. per square inch is equal to 46·22 feet of water-pressure, and by Table 11 the volume of 20-lb. steam is 732 for water 1, hence the height of the column of steam generating the velocity is $46{\cdot}22 \times 732 = 33833$ feet, and the velocity of efflux $\sqrt{33833} \times 8 = 1472$ feet per second.

(146.) It will be seen from this, that for finding the velocity of efflux in any case, we require only the difference of pressure at the two sides of the orifice and the density of the issuing gas or steam; this is further illustrated by (123), &c.

(147.) When the pressure varies *very slightly*, as is usually the case in most questions of ventilation, discharge of coal-gas, &c., we may admit, without sensible error, that the density is constant, and the velocity of discharge will then be governed by the square root of the pressure simply, and we have the rule $\sqrt{P} \times 66{\cdot}1 = V$ for air, and $\sqrt{P} \times 102 = V$ for coal-gas of density ·42 for air 1; P being the pressure in inches of water, and V = velocity in feet per second. Cols. 3 and 6 in Table 47 have been calculated by these rules. These rules may be modi-

TABLE 47.—Of the VELOCITY of DISCHARGE of COMMON AIR and of COAL-GAS, at ordinary Temperature and Pressure, with small differences of Pressure.

Head or Difference of Pressure in		Common Air at 62° under 30 inches of the Barometer.			Coal Gas, specific gravity ·42, that of Air being 1·0.		
		Coefficient of Discharge.			Coefficient of Discharge.		
Inches of Water.	Pounds per Square foot.	1·0	·93	·65	1·0	·93	·65
		Velocity in feet per second.			Velocity in feet per second.		
·005	·026	4·67	4·34	3·03	7·2	6·70	4·68
·01	·052	6·61	6·14	4·29	10·2	9·48	6·63
·02	·104	9·35	8·69	6·07	14·4	13·4	9·36
·03	·156	11·4	10·6	7·41	17·6	16·3	11·4
·04	·208	13·2	12·3	8·58	20·4	19·0	13·3
·05	·260	14·8	13·7	9·62	22·8	21·2	14·8
·07	·363	17·4	16·2	11·3	27·0	25·1	17·5
·10	·519	20·9	19·4	13·6	32·3	30·0	21·0
·15	·779	25·6	23·8	16·6	39·5	36·7	25·7
·2	1·038	29·5	27·4	19·2	45·6	42·4	29·6
·25	1·298	33·1	30·8	21·5	51·0	47·4	33·1
·3	1·558	36·2	33·6	23·5	55·8	51·9	36·2
·35	1·818	39·1	36·3	25·4	60·3	56·1	39·2
·4	2·077	41·8	38·8	27·2	64·5	60·0	41·9
·45	2·337	44·3	41·2	28·8	68·4	63·6	44·5
·5	2·597	46·7	43·4	30·3	72·1	67·0	46·8
·6	3·116	51·2	47·6	33·3	79·0	73·4	51·3
·7	3·635	55·3	51·4	35·9	85·4	79·4	55·5
·8	4·155	59·1	54·9	38·4	91·2	84·8	59·3
·9	4·674	62·7	58·3	40·7	96·8	90·0	62·9
1·0	5·193	66·1	61·4	42·9	102	95	66·3
1·5	7·790	80·9	75·2	52·5	125	116	81
2·0	10·38	93·5	86·9	60·7	144	134	94
2·5	12·98	104	96·7	67·6	161	149	104
3·0	15·58	114	106	74·1	176	163	114
3·5	18·18	124	115	80·6	191	177	124
4·0	20·77	132	123	85·8	204	190	133
4·5	23·37	140	130	91·0	216	201	140
5·0	25·97	148	138	96·2	228	212	148
6·0	31·16	162	151	105·3	250	232	162

fied so as to give the *quantity* discharged instead of the velocity. Thus for round apertures with air we have $d^2 \times \sqrt{P} \times 21 \cdot 64 = C$; and for coal-gas $d^2 \times \sqrt{P} \times 33 \cdot 4 = C$, in which C = cubic feet discharged per minute, d = diameter of orifice in inches, and P = pressure in inches of water.

(148.) "*Coefficient of Contraction.*"—But the quantities discharged will vary considerably with the form of the orifice, for the issuing vein of air, &c., suffers *contraction*, as in the case of water, and as shown by Fig. 34, where with a thin plate, an orifice 1 inch diameter has the jet reduced to ·8 diameter, and the area to $\cdot 8^2 = \cdot 64$, that of the orifice being 1. The experiments of Daubuisson give the following coefficients of discharge for air:—

·65 for an orifice in a thin plate.

·93 for a very short cylindrical pipe, say two diameters long.

·95 for a similar pipe slightly enlarged outward, or trumpet-shaped.

(149.) "*Velocity of Steam.*"—With steam of a fixed pressure (say 45 lbs. per square inch on one side of an orifice and 44 lbs. on the other side) we have a difference of 1 lb. per square inch, which by Table 41 is equal to 2·3 feet head of water; and by Table 11 we find the volume of 45 lbs. steam to be 439 times that of water, and hence the height of column of 45 lbs. steam, equivalent to 1 lb. pressure, is $2 \cdot 3 \times 439 = 1010$ feet; and the velocity due to that height is $\sqrt{1010} \times 8 = 254$ feet per second, which is the velocity at the most contracted part of the issuing jet (C, Fig. 34); but with a pipe taking ·93 for the co-efficient this becomes $254 \times \cdot 93 = 236$ feet per second over the full area of the orifice at B. From this we obtain the general rule $\sqrt{p} \times 236 = V$, in which p = the loss of pressure in pounds per square inch, and V = the velocity of the discharge over the full area of the orifice in feet per second. Table 48 has been calculated by these rules.

(150.) The discharge of steam may be sometimes conveniently estimated by the horse-power instead of cubic feet. Taking the case of 45 lbs. steam, and admitting that one cubic foot of water evaporated to steam represents 1-horse power (115), we find by Table 11 that a cubic foot of water is equal to 439 cubic feet of steam at that pressure, and to pass that quantity through an orifice 1 inch diameter we should require a velocity of $\frac{439 \times 144}{\cdot 7854 \times 60 \times 60} = 22 \cdot 36$ feet per second, which with a

TABLE 48.—Of the VELOCITY of DISCHARGE of 45 lbs. STEAM, with given differences of Pressure at the two sides of an orifice.

Loss of Pressure in Pounds per Square inch.	Co-efficient of Discharge.			Loss of Pressure in Pounds per Square inch.	Coefficient of Discharge.		
	1·0	·93	·65		1·0	·93	·65
	Velocity in feet per second.				Velocity in feet per second.		
·01	25·4	23·6	16·5	·6	197	183	128
·02	35·9	33·4	23·3	·7	213	198	138
·03	44·0	40·9	28·6	·8	228	211	148
·04	50·8	47·2	33·0	·9	241	224	157
·05	56·9	52·9	37·0	1·0	254	236	165
·1	80·4	78·4	52·3	1·1	267	248	173
·15	98·5	91·6	64·0	1·2	279	259	181
·2	114	106	74·1	1·3	290	270	188
·3	139	129	90·3	1·4	301	280	195
·4	161	150	105	1·5	311	289	202
·5	180	167	117				

pipe becomes $\frac{22 \cdot 36}{\cdot 93} = 24$ feet per second, the head due to which by the rule for falling bodies $\left(\text{namely } \left(\frac{V}{8}\right)^2 = H\right)$ becomes in our case $\left(\frac{24}{8}\right)^2 = 9$ feet head of steam, or $\frac{9}{439} =$ ·0205 feet of water, which again by Table 41 is equal to ·0205 × ·4327 = ·00887 lbs. pressure per square inch.

(151.) From this we get for circular apertures, or rather for *short* pipes discharging 45 lbs. steam, the general rule $\frac{H^2}{d^4} \times \cdot 00887 = p$, in which H = horse-power of steam, estimated in cubic feet of water; d = diameter of short pipe, say two diameters long, in inches; p = the difference of pressure at the two ends of the short pipe, in pounds per square inch. Thus, a short 3-inch pipe, say 6 or 8 inches long, discharging 120 horse-power of steam, would require $\frac{120^2}{3^4} \times 00887$, or $\frac{14400}{81} \times \cdot 00887 = 1 \cdot 57$ lbs. per square inch, so that the pressure at the exit end of the pipe is reduced from 45 to 45 − 1·57 = 43·43 lbs. per square inch. Table 49 has been calculated by this rule.

TABLE 49.—For 45 lbs. STEAM, giving the Loss of Pressure due only to Velocity of Entry into Steam-pipes, &c.

HORSE-POWER.			DIAMETER OF THE PIPE IN INCHES.									
With uniform Velocity.	Single Steam-engine.	Two Engines, Coupled Cranks at right angles.	1	1½	2	2½	3	4	5	6	7	8
			LOSS OF PRESSURE IN POUNDS PER SQUARE INCH.									
5	3·18	4·5	·222	·044	·0138	·0057	·0027					
10	6·37	9·0	·887	·175	·0554	·0227	·0109	·00347	·00142			
15	9·55	13·5	2·00	·394	·1246	·0511	·0245	·0078	·00319	·00154	·00083	
20	12·7	18·0	3·54	·700	·2216	·0908	·0436	·0139	·00568	·002736	·001476	·000868
30	19·1	27·0	..	1·575	·4986	·2043	·0981	·0312	·01278	·00615	·00332	·001953
40	25·5	36·0	..	2·800	·8864	·3652	·1744	·0555	·02272	·01094	·00590	·003472
50	31·8	45·0	..	4·375	1·385	·5675	·2725	·0867	·03550	·01710	·00922	·005425
75	47·7	67·6	..	..	3·116	1·276	·6131	·1952	·07987	·03847	·02075	·01220
100	63·7	90·1	..	..	..	2·270	1·09	·3470	·1420	·0684	·0369	·0217
150	95·5	135	..	..	..	..	2·45	·7807	·3190	·1539	·0830	·0488
200	127	180	..	..	..	..	..	1·388	·5680	·2736	·1476	·0868
250	159	225	..	..	..	..	..	2·168	·8875	·4275	·2306	·1356
300	191	270	..	..	..	..	..	..	1·278	·6150	·332	·1953
400	255	360	..	..	..	..	..	..	2·272	1·094	·590	·3472

For 7 lbs. steam, the rule becomes $\frac{H^2}{d^4} \times \cdot 0231 = p$; and by this we have calculated Table 50.

TABLE 50.—For 7 lbs. STEAM, giving the Loss of Pressure due only to Velocity of Entry into Steam-pipes.

HORSE-POWER.			DIAMETER OF THE PIPE IN INCHES.									
With uniform Velocity.	Single Steam-engine.	Two Engines, Coupled Cranks at right angles.	2	3	4	5	6	7	8	9	10	12
			LOSS OF PRESSURE IN POUNDS PER SQUARE INCH.									
10	6·37	9·0	·1444	·0285	·00900	·0037	·001783					
15	9·55	13·5	·3250	·0641	·0203	·0088	·00400	·00216				
20	12·7	18·0	·5776	·1140	·0361	·0148	·00712	·00385	·00225	·00141		
30	19·1	27·0	..	·2565	·0812	·0333	·01602	·00866	·00508	·00317	·00208	
40	25·5	36·0	..	·4560	·1443	·0592	·02848	·0154	·00902	·00563	·00370	·00178
50	31·8	45·0	..	·7125	·2255	·0925	·04450	·0241	·0141	·00880	·00578	·00277
75	47·7	67·6	..	..	·5073	·2164	·1001	·0540	·0317	·0198	·0130	·00624
100	63·7	90·1	..	..	·9020	·3700	·1783	·0962	·0564	·0352	·0231	·0111
150	95·5	135	..	..	..	·8320	·4005	·1948	·1269	·0792	·0520	·0250
200	127	180	..	..	..	..	·7120	·385	·225	·1410	·0924	·0444
300	191	270	..	..	..	..	..	·866	·508	·3170	·208	·1000
400	255	360	..	..	..	..	..	..	·902	·5630	·370	·178
500	318	450	..	..	.	..	..	..	..	·8800	·578	·277

(152.) The pressure varies as the fourth power of the diameter of the orifice, as shown by the preceding rules, for this reason :—the area of an orifice, and consequently the velocity of efflux with a given quantity, varies as d^2; and as the pressure varies as V^2, the ratio becomes $d^2 \times V^2$, or more simply as d^4. For instance, if in any particular case we reduce the diameter of an orifice to one half, the velocity would be increased in the ratio of 1 to 4, and the pressure from 1 to 16.

(153.) "*Friction of Long Pipes.*"—We have so far considered only the head or pressure necessary to give any required velocity with an orifice, or a pipe so short that the friction was inappreciable; but where the length is considerable, there is a second loss of pressure due simply to the *friction* of the air or steam against the sides of the pipe; and in all such cases the head due both to velocity and to friction must be separately calculated and the sum total taken.

(154.) The head due to friction alone may be calculated by the following rules—

$$\frac{C^2 \times L}{(3 \cdot 7d)^5} = H, \frac{(3 \cdot 7d)^5 \times H}{L} = C^2, \frac{C^2 \times L}{H} = (3 \cdot 7d)^5, \frac{(3 \cdot 7d)^5}{C^2} \times H = L,$$

in which d = the diameter of the pipe in inches, L = the length of pipe in yards, C = cubic feet per minute, and H = the head (or difference of pressure at the two ends of the pipe) in inches of water. These rules cannot be easily worked without the use of logarithms; but we have calculated by them Table 51, for the use of which we have the following rules :—

(155.) 1st. Having the diameter, length, and discharge given, to find the head, take from Table 51, opposite the given diameter, the number in column 2 or 3 (according to the terms in which the head is desired), and multiply it by the square of the given discharge in cubic feet, and by the length in yards; and the product is the head in inches of water, or in pounds per square inch, due to friction alone, to which the head due to velocity has to be added from Tables 47 or 48, &c.

(156.) 2nd. Having the head, diameter, and length given, to find the discharge, assume a discharge and calculate the head for that as in (155); divide the assumed discharge by the square root of the head due to it, and multiply by the square root of the given head; and the quotient is the true discharge sought.

(157.) In order to facilitate calculation, the Table 51 is so arranged, that for diameters under one inch the discharge must

TABLE 51.—Of the FRICTION of AIR, STEAM, and GAS in LONG PIPES.

Diameter of Pipe in inches.	Head, or difference of Pressure at the two ends of a Pipe one yard long, in	
	Inches of Water.	Pounds per Square Inch.
	Head for 1 Cubic Foot per Minute.	
¼	1·477	·05317
⅜	·1945	·00700
½	·0185	·000666
¾	·006077	·0002187
1	·001442	·0000519
	Head for 10 Cubic Feet per Minute.	
1¼	·04725	·001701
1½	·01899	·000684
1¾	·00878	·000326
2	·00451	·000162
2¼	·00250	·000090
2½	·00148	·0000532
	Head for 100 Cubic Feet per Minute.	
3	·0594	·00214
3½	·0746	·000989
4	·0141	·000507
4½	·00782	·000281
5	·00462	·000166
6	·00186	·0000668
7	·000858	·0000309
8	·000440	·0000158
	Head for 1000 Cubic Feet per Minute.	
9	·02442	·000879
10	·01442	·000519
12	·00579	·000209
15	·00189	·0000681
18	·000763	·0000275
21	·000353	·0000127
24	·000181	·00000653
	Head for 10,000 Cubic Feet per Minute.	
30	·00593	·000214
33	·00368	·000133
36	·00238	·0000858
42	·001103	·0000398
48	·000566	·0000204
54	·000314	·0000113
60	·000185	·00000667

be taken in cubic feet, as in (155), &c., but from 1¼ inch to 2½ inches in tens, from 3 inches to 8 inches in hundreds, from 9 inches to 24 inches in thousands, and from 30 to 60 inches in tens of thousands of cubic feet.

Thus a ¾-inch pipe, 20 yards long, discharging 2 cubic feet of air per minute, requires $\cdot006 \times 2^2 \times 20 = \cdot48$ inch head of water for friction, and the velocity being $\frac{144 \times 2}{\cdot44 \times 60} = 10\cdot9$ feet per second, the head for that velocity by Table 47 is $\cdot03$ inch of water, making $\cdot48 + \cdot03 = \cdot51$ inch total.

A 2½-inch pipe, 40 yards long, with 38 cubic feet of air, will require $\cdot00148 \times \left(\frac{38}{10}\right)^2 \times 40 = \cdot85$ inch head for friction, and the velocity being $\frac{144 \times 38}{4\cdot9 \times 60} = 18\cdot6$ feet per second, or by Table 47 $= \cdot1$ inch head, we have a total of $\cdot85 + \cdot1 = \cdot95$ inch head.

A 4-inch pipe, 10,560 yards long, discharging 852 cubic feet of *gas* per hour, or $14\cdot2$ cubic feet per minute, will require $\cdot0141 \times \left(\frac{14\cdot2}{100}\right)^2 \times 10{,}560 = 3\cdot002$ inches head for friction. With so small a discharge, the head for velocity will be inappreciable; Clegg found by experiment, that the head with a pipe having these conditions was 3 inches of water.

A 9-inch pipe, 30 yards long, discharging 500 cubic feet of gas per minute, will require $\cdot02442 \times \left(\frac{500}{1000}\right)^2 \times 30 = \cdot183$ inch head for friction, and the velocity being $\frac{144 \times 500}{63\cdot6 \times 60} = 18\cdot8$ feet per second, the head for that velocity by Table 47 is $\cdot04$ inch, or $\cdot183 + \cdot04 = \cdot223$ inch total.

(158.) Again, say we require to know the discharge of 45 lbs. steam by a 4-inch pipe, 150 yards long, with a loss of 1 lb. in pressure, the steam at the exit end being reduced to 44 lbs. per square inch. Let us assume, say 400 cubic feet for the discharge, and calculating as in (156), we have $\cdot000507 \times \left(\frac{400}{100}\right)^2 \times 150 = 1\cdot22$ lb. per square inch, as the head due to friction alone, the velocity will be $\frac{144 \times 400}{12\cdot6 \times 60} = 76\cdot2$ feet per second, which, by Table 48, with $\cdot93$ co-efficient, requires about $\cdot1$ lb. per square inch, so that for our ***assumed*** quantity, the total head is $1 \cdot 22 + \cdot 1 = 1 \cdot 32$ lb. per square inch, instead of 1 lb., the pressure allowed in our case. The true discharge (156) with the given head is therefore $\frac{400}{\sqrt{1 \cdot 32}} \times \sqrt{1} = 348$ cubic feet of steam per minute.

(159.) This last result is obtained by the application of a useful general law, which may be stated thus: the discharge of any pipe or system of pipes, apertures, &c., &c., is proportional to the square root of the head; and conversely, the head is proportional to the square of the discharge.

(160.) "*Square and Rectangular Channels.*"—The case of square and other rectangular channels may be assimilated to that of round pipes, and then the velocity, &c., may be calculated by the rules and tables given for the latter. The velocity of discharge, whatever may be the form of the pipe or channel, is in all cases proportional to the sectional area divided by the periphery or circumference. In round pipes, this is always one-fourth of the diameter: thus a pipe 1 inch diameter has an area of $\cdot 7854$, and a circumference of $3 \cdot 142$, and $\frac{\cdot 7854}{3 \cdot 142} = \cdot 25$; and a 4-inch pipe gives $\frac{12 \cdot 57}{12 \cdot 57} = 1$.

"*Square Pipes.*"—Square pipes give the same uniform ratio: thus a pipe one inch square will have an area of 1, and a periphery of 4 inches, and $\frac{1}{4} = \cdot 25$ as with a circular pipe: again, a pipe 4 inches square has an area of 16 square inches, and a periphery of 16 inches also, and as with the round pipe $\frac{16}{16} = 1$, it follows that the *velocity* of discharge with a square pipe is the same as with a round one, with the same length and head, &c.; but of course the quantity discharged will be greater with a square pipe in the same proportion, as the area of a square is greater than that of a circle, or as 1 to $\cdot 7854$.

"*Rectangular Pipes.*"—The same laws apply to rectangular pipes: thus a pipe 6 inches by 3 inches has an area of 18, and a periphery of 18 also, and $\frac{18}{18} = 1$, which is the same as a 4-inch pipe, as we have seen; therefore a round pipe 4 inches diameter will have the same velocity of discharge as a pipe 4 inches square, or as another 6 in. × 3 in., and the quantities discharged will be in proportion to the areas, or $12 \cdot 57$, 16, and 18 respectively.

Say we have a rectangular channel, 40 yards long, 36 inches wide, and 18 inches deep, and we require to know the head or pressure for a velocity of 6 feet per second. The area is 36 × 18 = 648, and the periphery 108, and we have $\frac{648}{108} = 6$, which is the same as a pipe 6 × 4 = 24 inches diameter, and we can calculate the head as for a round pipe of that diameter. A round 24-inch or 2-feet pipe, discharging at the rate of 6 feet per second, would deliver $3 \cdot 14 \times 6 \times 60 = 1130$ cubic feet per minute, requiring $\cdot 000181 \times \left(\frac{1130}{1000}\right)^2 \times 40 = \cdot 0077$ inch head for friction alone. The head for 6 feet velocity by Table 47 with ·93 for co-efficient, is ·01 inch of water; the total head is therefore $\cdot 0077 + \cdot 01 = \cdot 0177$ inch of water, and this is also the head for a pipe 36 inches by 18 inches, as in our case.

(161.) "*Effect of Repeated Enlargements and Contractions.*"—It might be supposed, that the effect of enlarging the channel would be to diminish friction in discharging a fixed quantity of air, and this is true where the velocity has not to be got up again; but where there are repeated and successive contractions and enlargements, the head saved in friction by each enlargement, may be more than compensated by that required for getting up the velocity at the next contraction. Let Fig. 113 represent the rectangular pipe 40 yards long, the head for which with a velocity of 6 feet per second we have just calculated to be ·0177 inches of water.

Let Fig. 114 be a similar pipe, but one having two chambers or rooms, A and B, in its course, by which the length of the *pipe* or channel itself, C F, is reduced to one-half or 20 feet. The velocity of the air passing through the two rooms is so very small, that there will practically be no friction there, so that the friction is thus reduced to one-half also. But we found that the head at C, due to the air entering the pipe with 6 feet velocity, was ·01; when the air enters the room A, that velocity is lost, and must be got up again for the air entering the next contraction at D, to be again lost in B, and got up again at E. The head for friction in this case is $\frac{\cdot 0077}{2} = \cdot 00385$, and the head for the velocity at 3 places $\cdot 01 \times 3 = \cdot 03$, making a total of $\cdot 00385 + \cdot 03 = \cdot 03385$, or about double the head for a uniform pipe; so far therefore from diminishing the head by enlarging the channel at A and B, we have really doubled it in

TABLE 52.—Of the FRICTION of STEAM-PIPES for 45 lbs. STEAM, being the head lost by a length of one yard of Pipe.

HORSE-POWER.			DIAMETER OF THE PIPE IN INCHES.									
With uniform Velocity.	Single Steam-engine.	Two Engines, Coupled Cranks at right angles.	1	1½	2	2½	3	4	5	6	7	8
			PRESSURE, OR HEAD, LOST IN POUNDS PER SQUARE INCH.									
5	3·18	4·5	·069	·0091	·00216	·00071						
10	6·37	9·0	·277	·0365	·00866	·00284	·00114					
15	9·5	13·5	·623	·0821	·01948	·0639	·00256	·000608				
20	12·7	18·0	1·108	·1460	·03464	·01136	·00456	·00108	·0003548			
30	19·1	27·0	2·493	·3285	·07794	·02556	·01026	·00243	·0007983	·000322		
40	25·5	36·0	..	·5840	·1386	·04544	·01824	·00432	·00142	·000571	·000264	
50	31·8	45·0	..	·9125	·2165	·07100	·0285	·00675	·00222	·000893	·000413	·000212
75	47·7	67·6	..	2·052	·487	·1597	·064	·0152	·0050	·00200	·00092	·000475
100	63·7	90·1	..	..	·866	·284	·114	·027	·0089	·00357	·00165	·000846
150	95·5	135	..	..	1·95	·639	·256	·061	·0200	·0080	·0037	·0019
200	127	180	..	..	..	1·136	·456	·108	·0355	·0143	·0066	·00338
250	159	225	..	..	..	..	·713	·169	·0554	·0223	·0103	·00529
300	191	270	..	..	..	..	1·026	·243	·0798	·0322	·0148	·00761
400	255	360	..	..	..	..	..	·432	·142	·0571	·0264	·01352

TABLE 53.—Of the FRICTION of 7 lbs. STEAM passing through PIPES one yard long.

HORSE-POWER.			DIAMETER OF THE PIPE IN INCHES.									
With uniform Velocity.	Single Steam-engine.	Two Engines, Cranks at right angles.	2	3	4	5	6	7	8	9	10	12
			PRESSURE, OR HEAD, LOST IN POUNDS PER SQUARE INCH.									
10	6·37	9·0	·0587	·00774	·00184	·000602						
15	9·5	13·5	·1321	·01741	·00414	·00136	·00054					
20	12·7	18·0	·2348	·03096	·00736	·00241	·00097	·00045				
30	19·1	27·0	·5283	·06966	·01656	·00542	·00218	·00101	·00052			
40	25·5	36·0	..	·1238	·02944	·00963	·00387	·00179	·00092	·00051		
50	31·8	45·0	..	·1935	·046	·0151	·00605	·00280	·00144	·00079	·00047	
75	47·7	67·6	..	·435	·103	·0340	·0136	·0063	·00325	·00174	·00105	·000425
100	63·7	90·1	..	·774	·184	·0602	·0242	·0112	·00574	·00318	·00188	·000756
150	95·5	135	..	..	·414	·136	·0545	·0252	·0129	·00715	·00422	·00170
200	127	180	..	..	·736	·241	·0968	·0448	·0230	·0127	·00752	·00302
300	191	270	..	..	..	·542	·218	·1008	·0517	·0286	·0169	·00680
400	255	360	..	..	..	..	·387	·179	·0918	·0509	·0301	·0121
500	318	450	..	..	..	..	..	·280	·144	·0795	·0470	·0189

this case. It is important to keep this fact in view in cases of ventilation, &c.; in large buildings, the changes of area in the passages are numerous and unavoidable, they are also too complicated to be calculated, but in the case of the Prison Mazas (362), we found that the velocity was thus reduced to ·423 of the theoretical velocity, and in the Prison of Provins (365), to ·322.

(162.) "*Steam Pipes.*"—Representing 1-horse power by one cubic foot of water evaporated to steam, we have, by Table 11, for 45 lbs. pressure, 439 cubic feet of steam per horse-power per hour, or 7·3 cubic feet per minute, and for a pipe 1 inch diameter, 1 yard long, and 10-horse power, we have by the rule $\frac{C^2 \times L}{(3 \cdot 7d)^5} = H$, in our case, $\frac{(7 \cdot 3 \times 10)^2 \times 1}{(3 \cdot 7 \times 1)^5} = 7 \cdot 685$ inches head of water, or $\frac{7 \cdot 685}{12} \times \cdot 4327 = \cdot 277$ lb. per square inch, and thus we have calculated Table 52.

With 7 lbs. steam we have by Table 11, 1138 cubic feet of steam per horse-power per hour, or 19 cubic feet per minute, and for 10-horse power and a pipe 1 inch diameter we require $\frac{190^2 \times 1}{(3 \cdot 7 \times 1)^5} = 52 \cdot 06$ inches head of water, or $\frac{52 \cdot 06 \times \cdot 4237}{12} = 1 \cdot 88$ lb. per square inch, as in Table 53.

(163.) "*Steam-Pipes to Engines.*"—In applying these rules to pipes for steam-engines, it must be observed that the supply of steam to an ordinary engine is intermittent, being 0 when passing the centre, and a maximum when at the middle of the stroke, and that the maximum velocity should be taken in calculating the size of steam-pipe. Thus, an engine of 5-feet stroke and 22 revolutions per minute has a *mean* velocity of 5 × 2 × 22 = 220 feet per minute; but when at the centre of the stroke, the piston is moving with the velocity of the crank pin, or 5 × 3·14 × 22 = 345 feet per minute, instead of 220. It will be found that in a common double-acting engine the maximum velocity is 1·57 times the mean velocity, and with a pair of engines having their cranks at right angles the ratio is 1·11 to 1: so that, for instance, one engine of 100-horse power takes steam at the maximum rate of 157-horse power, and a pair of engines, each of 50-horse power, combined at right angles, takes 111-horse power of steam at the maximum speed. From this we have columns 2 and 3 in Tables 49, 52, &c.

(164.) The following examples will illustrate the use of these Tables:—

Example 1. A single engine of 25-horse power, with a 2½-inch steam-pipe, 50 yards long and 45 lbs. steam, will by Table 49 require ·365 lb. pressure for velocity of entry, and by Table 52 ·0454 lb. per yard for friction, or in our case ·0454 × 50 = 2·27 lbs. per square inch, making a total of ·365 + 2·27 = 2·635 lbs., so that the available pressure for working the engine is reduced to 45 − 2·635 = 42·365 lbs. per square inch.

Example 2. A single low-pressure engine of 250-horse power, with 7 lbs. steam, 10-inch pipe, 20 yards long, will require by Table 53 (say 255-horse power) ·0301 × 20 = ·602 lb. for friction, and by Table 50, 37 lbs. for velocity, making a total loss of ·602 + ·37 = ·972 lb. per square inch, thus reducing the effective pressure at the engine to 7 − ·972 = 6·028 lbs. per square inch.

Example 3. A pair of engines, of the collective power of 180 horses, with a 5-inch pipe, 40 yards long and 45 lbs. steam, would require by Table 52, ·0355 × 40 = 1·42 lb. for friction, and by Table 49, ·568 for velocity, giving a total loss of 1·42 + ·568 = 1·988—say 2 lbs. per square inch.

Example 4. Say that the steam of a 20-horse boiler is used for evaporating pans or similar work where the velocity is uniform, then with 45 lbs. steam and 100 yards of 2-inch pipe, we should have a loss of (·03464 × 100) + ·2216 = 3·6856 lbs. per square inch.

(165.) With steam of other pressures than 45 or 7 lbs. per square inch, the pressure lost must be calculated by Table 51, &c. Thus, say we have an engine with a 15-inch cylinder, 3 feet stroke, making 40 revolutions per minute, and that the steam-pipe is 3 inches diameter and 30 yards long, the pressure of steam in the boiler being 25 lbs. per square inch. The maximum velocity of the piston is 3 × 3·14 × 40 = 377 feet per minute, or 6·3 feet per second, which with a 3-inch pipe becomes $\frac{6 \cdot 3 \times 15^2}{3^2} = 157$ feet per second, or with ·93 for coefficient, $\frac{157}{\cdot 93} = 168$ feet per second, in the contraction at entry (148), the head due to which is $\left(\frac{168}{8}\right)^2 = 441$ feet of steam, and the volume of 25 lbs. steam, being by Table 11 = 644 times that of water, we have $\frac{441}{644} = \cdot 684$ feet of water, or $\cdot 684 \times \cdot 4327 = \cdot 296$ lb. per square inch, as the pressure due to velocity only.

Then the area of the piston being 1·227 feet, we have $1 \cdot 227 \times 377 = 462$ cubic feet of steam per minute as the maximum quantity passing through the pipe; and by Table 51 we shall have for friction alone $\cdot 00214 \times \left(\frac{462}{100}\right)^2 \times 30 = 1 \cdot 38$ lb. per square inch, which added to the head due to velocity makes the total loss of pressure $\cdot 296 + 1 \cdot 38 = 1.676$ lb. per square inch, and the effective pressure at the engine is reduced to $25 - 1 \cdot 676 = 23 \cdot 324$ lbs. per square inch.

CHAPTER V.

CHIMNEYS.

(166.) The case of a chimney is analogous to that in (141) and (145), where we had two columns of equal height but unequal density, and we found that the velocity was that due to the reduced difference of the two columns. It has been shown in (93) that the temperature of the air, &c., in well-arranged steam-boiler chimneys is about 550°, and by Table 19 it will be found that at that temperature the density of air is almost exactly one-half of the density at 62°, and we have, as in Fig. 35, a column of air in the chimney, say 80 feet high, weighing only half as much as the same height of external air, and motion will ensue as in (145) with a velocity due to $80 - 40 = 40$ feet head. For the purposes of calculation this head may be represented in inches of water, thus the density of air at 62° being $\frac{1}{820}$th that of water, and at 552° $\frac{1}{1640}$th a column 80 feet or 960 inches high will be equal to $\frac{960}{820} = 1 \cdot 17$ and $\frac{960}{1640} = \cdot 585$ inch respectively, and the difference is $1 \cdot 17 - \cdot 585 = \cdot 585$ inch of water, see (221).

Assuming a fixed or standard temperature for the chimney at 552°, we have in Table 54 the equivalent differences of pressure in inches of water, by which we may calculate the velocities, &c., by the rules and Tables 47, 51, &c., &c.

TABLE 54.—Of the DRAUGHT POWERS of CHIMNEYS, &c., with the Internal Air at 552°, and External Air at 62°, and with the Damper nearly closed.

Height of Chimney in Feet.	Draught Power in Inches of Water.	Theoretical Velocity in Feet per Second.		Height of Chimney in Feet.	Draught Power in Inches of Water.	Theoretical Velocity in Feet per Second.	
		Cold Air entering.	Hot Air at exit.			Cold Air entering.	Hot Air at exit.
10	·073	17·8	35·6	80	·585	50·6	101·2
20	·146	25·3	50·6	90	·657	53·7	107·4
30	·219	31·0	62·0	100	·730	56·5	113·0
40	·292	35·7	71·4	120	·876	62·0	124·0
50	·365	40·0	80·0	150	1·095	69·3	138·6
60	·438	43·8	87·6	175	1·277	74·8	149·6
70	·511	47·3	94·6	200	1·460	80·0	160·0

We allowed in (90) 300 cubic feet of air at 62° per pound of coal; in passing through the fire this is highly heated and it leaves at 1279° (89) and is expanded to about 3½ times its former volume (Table 19); from thence to the chimney it is progressively cooled to 552°, and becomes reduced to double its normal volume, or to 600 cubic feet. If we allow 10 lbs. of coal per horse-power, we have to pass 6000 cubic feet up the chimney per horse-power per hour.

(167.) "*Round Chimneys.*"—Say we require the power of a chimney 80 feet high, 2 feet 9 inches diameter, attached to steam-boilers 30 feet long, having flues the same area as the chimney, and say 100 feet long in circuit from furnace to chimney. It will be seen that we have to determine the discharge of a pipe 180 feet or 60 yards long, 2 feet 9 inches diameter, with a head of ·585 inch of water by Table 54 and (166). We must assume a discharge as in (156), say 100-horse power or $\frac{100 \times 6000}{60}$ = 10000 cubic feet per minute, which will require by the rule in (154) $\frac{10000^2 \times 60}{(3 \cdot 7 \times 33)^5}$ = ·2211 inch of water for friction alone; we have to add to this the head due to velocity. The diameter being 2·75 feet, we have an area of 5·94 feet, and as we have $\frac{10000}{60}$ = 167 cubic feet per second, the velocity will be $\frac{167}{5 \cdot 94}$ = 28 feet per second, which by Table 47, with ·93 coefficient, is due to a head of $\cdot 2$ inch of water, and the total head for 100-horse power is $\cdot 2211 + \cdot 2 = \cdot 4211$ inch of water. We have, however, $\cdot 585$ inch at disposal, and by (156) or (159) this will be equal to $\frac{100 \times \sqrt{\cdot 585}}{\sqrt{\cdot 4211}}$ or $\frac{100 \times \cdot 765}{\cdot 649} =$ 118-horse power. A chimney of these dimensions is working well at Dartford, the consumption of coal being 10 cwt. per hour, which allowing 10 lbs. per horse-power, as we assumed in (166), is equal to $\frac{1120}{10} =$ 112-horse power.

(168.) "*Square Chimneys.*"—If the chimney we have just considered had been square, the horse-power would have been greater in the simple proportion of the areas of a square to a circle, see (160). The *velocity* of discharge is the same in both cases, and the quantity discharged is proportional to the respective areas, or as $\cdot 7854$ to 1; in our example (167), a square chimney would have been equal to $\frac{118}{\cdot 7854} =$ 150-horse power.

(169.) We shall assume a constant length of circuit of flues at 100 feet, this will be too great for small boilers, but the only effect will be to make the chimney rather too powerful for such cases. It is expedient to allow a margin for unforeseen contingencies, and to give an excess of power in all cases; the damper can be regulated so as to obviate any mischievous results from the admission of an excess of air (101). We have calculated, Table 55, on these principles, giving the power throughout at 75 per cent. of the maximum calculated power, thus allowing 25 per cent. for margin.

Figs. 44 to 46 give elevations of common chimneys of 40, 60, and 80 feet height; care should be taken not to contract the channel at the points B, C, D, to less area than the outlet A at the top. Mortar should be used for the most part, because cement is destroyed by a strong heat, the 4½-inch work at the top, however, should be in good cement; with so thin a wall the heat is rapidly carried off by the external air, and the cement will not be injured. With steam-boilers the heat of the air should not exceed 600° (93), and ordinary stock-bricks will stand that temperature well, but with reverberatory and other brick furnaces (78), the air is at a temperature of about 2250°, and for such cases the chimney should be lined with fire-brick throughout, and as the cohesion of mortar is soon destroyed with such high temperatures, there should be wrought-iron bands

TABLE 55.—Of the POWER of CHIMNEYS to STEAM-BOILERS, having flues 100 feet long, in circuit from Furnace to Chimney.

Size at the Top, inside.	40 feet.		60 feet.		80 feet.		100 feet.		120 feet.		150 feet.	
	Round.	Square.	Round.	Square.	Round.	Square.	Round.	Square.	Round.	Square.	Round.	Square.
ft. in.	H.P.	H.P.										
1 0	6·4	8·1										
1 3	10·9	13·9	12·8	16·3								
1 6	16·6	21·0	19·5	24·8	21·7	27·5						
1 9	23·6	30·0	27·9	34·2	31·1	40·0						
2 0	31·9	41·0	37·3	47·5	42·3	53·8	45·7	58·2				
2 3	..	..	49·4	62·8	55·3	70·4	60·0	76·4	63·8	81·2		
2 6	..	..	65·3	83·1	70·4	90·0	76·5	97·4	81	103	85	108
2 9	..	..	78	100·0	88	112	94·9	121	101	128	106	135
3 0	..	..	94	123	106	135	114	145	123	157	130	165
3 6	..	..	..	..	150	191	163	207	175	223	186	237
4 0	..	..	..	..	202	257	220	280	235	300	252	321
5 0	..	..	..	..	..	..	360	458	388	494	415	528
6 0	..	..	..	..	..	..	..	..	577	734	615	783

NOTE.—The power of the chimneys in this Table is three-fourths of their absolute maximum power: thus the maximum power of a chimney 3 ft. 6 in. diameter, 80 feet high, is $\frac{150 \times 4}{3}$ = 200-horse power, &c.

round the outside at regular distances from top to bottom. In ordinary chimneys hoop-iron should be built into the brickwork every few courses to form a bond; and a lightning conductor should not be omitted.

(170.) "*Effect of Long and Short Flues.*"—The effect of different lengths of flue is shown by Table 56, in which we have taken as an example a chimney 60 feet high and 2 feet 9 inches square, which by Table 55 with an ordinary flue 100 feet long, is equal to 100-horse power; it will be seen that with a flue of one-half the length, or 50 feet, the power is increased to 107·6-horse power only, and that with a flue 1000 feet long, the power is reduced to one-half nearly. This may be applied to other cases, say we required a chimney of 150-horse power, with a flue 1000 feet long (from furnace to chimney), this would be equal to $\frac{150}{\cdot 51}$ = 300-horse power in Table 55, and might be 120 feet high and 4 feet square. Again, a chimney of 50-horse power, with a flue 400 feet long, must be equal to $\frac{50}{\cdot 708}$ = 70-horse power, in Table 55, and may be 80 feet high and either 2 feet 6 inches round, or 2 feet 3 inches square, &c., &c.

TABLE 56.—Of the POWER of a CHIMNEY 60 ft. high, 2 ft. 9 in. square, with Flues of different Lengths.

Length of Flue in Feet.	Horse-power.	Length of Flue in Feet.	Horse-power.
50	107·6	800	56·1
100	100·0	1000	51·4
200	85·3	1500	43·3
400	70·8	2000	35·9
600	62·5		

(171.) Péclet has given rules for the calculation of chimneys, by which we have calculated Table 57; but we have certainly known chimneys doing nearly double the work assigned by that Table, for instance the chimney at Dartford (167) would be by this Table about 60-horse power, whereas our calculation gives its *maximum* power at 118 horses, and as we have shown it is really doing 112-horse power satisfactorily.

Table 57.—Of the Power of Chimneys to Steam Boilers, according to the rules of Péclet.

Horse-power of Chimney.		Assumed Length of circuit of Flues in Feet.	Height of the Chimney in Feet.						
Square.	Round.		20	40	60	80	100	125	150
			Minimum Size of Chimney in Inches.						
10	7·8	45	18·40	14·12	12·95				
20	16	65	23·30	19·73	18·12	17·07			
35	27	85	30·73	26·10	23·83	22·39			
50	39	105	36·76	31·17	28·41	26·06	25·45		
75	59	110	..	37·96	34·56	32·39	30·88		
100	78	110	..	..	39·63	37·11	35·35	34·52	
150	118	110	..	..	48·34	44·19	43·03	41·00	39·57
200	157	110	..	..	55·55	51·95	49·40	47·04	45·38
300	235	120	..	..	..	..	60·23	57·33	55·20
400	315	130	..	..	..	..	..	65·97	63·48
500	390	150	..	..	..	..	..	..	70·85

See Table 55, which agrees better with experience.

CHAPTER VI.

ON VAPOURS.

(172.) Let Fig. 57 be a barometer with a large chamber A, one cubic foot in capacity, and let the bore of the tube be very small, so that its capacity may be regarded as infinitely small compared with that of the chamber; the space A will then be a perfect vacuum and the height of the column of mercury at ordinary atmospheric pressure will be say 30 inches. Now if a few drops of water be introduced into A, vapour will instantly be formed from it, filling the chamber and depressing the column B say to G, the amount of this depression will depend on the temperature of A; by adding mercury at C until that column is raised say to D, the column B may be restored to its former level. Now the height of the column C B was the measure of the atmospheric pressure, and as that is supposed to be constant, it is evident that a pressure equal to C D is exerted by the vapour in the chamber A, thus if C D is 1 inch, D B will then be 29 inches only, and as 30 is required to balance the atmospheric pressure, the rest must have been made up by the pressure of the vapour in A, and is equal in this case to 1 inch of mercury. The experiments of Regnault give the elasticities or pressure of vapour of water at different temperatures as in Table 12 and in column 4 of Table 58, thus at 212° the boiling-point of water, column C, would have been raised to E, at the same level as B, showing that at that temperature the elastic force of vapour is equal to that of the atmosphere.

(173.) The weight of water contained in the vapour may now be calculated; the experiments of Regnault and Despretz have shown that the weight of a given volume of vapour of water is ·623, or nearly $\frac{5}{8}$ the weight of the same volume of air at the same temperature and pressure. (See Table 32.) Thus at 132°, the weight of a cubic foot of dry air at atmospheric pressure is given by column 3 of Table 58 at ·0671 lb., but the force of vapour at 132° is 4·752 inches of mercury, under which pressure air would weigh $\frac{\cdot 0671 \times 4\cdot 752}{29\cdot 921} = \cdot 010656$ lb. only, therefore a cubic foot of vapour will be $\cdot 010656 \times \cdot 623 = \cdot 006639$ lb., and thus has been calculated column 7 of Table 58.

(174.) By opening the cock at F we may admit the atmospheric air; if the space A, Fig. 57, had been a vacuum we should of course have then a cubic foot of dry air there, whose weight is given by column 2 in Table 58, but vapour being present, less than a cubic foot will be required. Say we take the case in which the elastic force of the vapour is 15 inches, or about half that of the atmosphere. If we assume that the relative volumes and densities of vapour follow the law of Marriotte (15) as dry air and gases do (which is not a fact, but the correctness of our deduction will not be affected in this case), we shall have a cubic foot of vapour at 15 inches pressure or elastic force, and if a piston were fitted into the cube, as at Fig. 58, and forced to descend from A to B, the pressure would increase from 15 inches at A to 30 inches at B, and the cubic foot of vapour would of course be reduced to half a cubic foot, its density being doubled and its elastic force increased to that of the atmosphere, also the space C above the piston would be a vacuum; if now the cock F were opened the air would fill the vacant half cubic foot, and the piston being removed the two would intermix with one another, but the relative weights would not be altered, the vapour would return to its normal volume and pressure of 15 inches of mercury, and the air being dilated

TABLE 58.—Of the WEIGHT of AIR, VAPOUR of WATER, and MIXTURES of AIR SATURATED with VAPOUR, at different Temperatures, under the ordinary Atmospheric Pressure of 29·921 inches in the Barometer.

Temp. Fahr.	Volume of Dry Air at different Temperatures, the Volume at 32° being 1·000.	Weight of a Cubic foot of Dry Air at different Temperatures in Pounds.	Elastic Force of Vapour in inches of Mercury. Regnault.	MIXTURE OF AIR SATURATED WITH VAPOUR.						Cubic feet of Vapour from 1 lb. of Water at its own Pressure in Column 4.
				Elastic Force of the Air in the Mixture of Air and Vapour in inches of Mercury.	Weight of a Cubic foot of the Mixture of Air and Vapour.			Weight of Vapour in 1 lb. of Air, in Pounds.	Weight of Dry Air mixed with 1 lb. of Vapour, in Pounds.	
					Weight of the Air in Pounds.	Weight of the Vapour in Pounds.	Total Weight in Pounds.			
32°	1·000	·0807	·181	29·740	·0802	·000304	·080504	·00379	263·81	3289
42	1·020	·0791	·267	29·654	·0784	·000440	·078840	·00561	178·18	2252
52	1·041	·0776	·388	29·533	·0766	·000627	·077227	·00819	122·17	1595
62	1·061	·0761	·556	29·365	·0747	·000881	·075581	·01179	84·79	1135
72	1·082	·0747	·785	29·136	·0727	·001221	·073921	·01680	59·54	819
82	1·102	·0733	1·092	28·829	·0706	·001667	·072267	·02361	42·35	600
92	1·122	·0720	1·501	28·420	·0684	·002250	·070717	·03289	30·40	444
102	1·143	·0707	2·036	27·885	·0659	·002997	·068897	·04547	21·98	334
112	1·163	·0694	2·731	27·190	·0631	·003946	·067046	·06253	15·99	253
122	1·184	·0682	3·621	26·300	·0599	·005142	·065042	·08584	11·65	194
132	1·204	·0671	4·752	25·169	·0564	·006639	·063039	·11771	8·49	151
142	1·224	·0660	6·165	23·756	·0524	·008473	·060873	·16170	6·18	118
152	1·245	·0649	7·930	21·991	·0477	·010716	·058416	·22465	4·45	93·3
162	1·265	·0638	10·099	19·822	·0423	·013415	·055715	·31713	3·15	74·5
172	1·285	·0628	12·758	17·163	·0360	·016682	·052682	·46338	2·16	59·2
182	1·306	·0618	15·960	13·961	·0288	·020536	·049336	·71300	1·402	48·6
192	1·326	·0609	19·828	10·093	·0205	·025142	·045642	1·22643	·815	39·8
202	1·347	·0600	24·450	5·471	·0109	·030545	·041445	2·80230	·357	32·7
212	1·367	·0591	29·921	0·000	·0000	·036820	·036820	infinite	·000	27·1
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)

to double its normal volume and reduced to half its normal density and pressure, or to 15 inches of mercury also.

(175.) In all cases the *sum* of the elastic forces of the vapour and air will be equal to the atmospheric pressure, the vapour taking always the force due to its temperature as per column 4 in Table 58, and the air making up the complement.

(176.) Returning to our former illustration;—at 132° the vapour had a force of 4·752 inches, the pressure of the air in that case therefore must be 29·921 − 4·752 = 25·169 inches and a cubic foot of air at that density would weigh $\frac{\cdot 0671 \times 25\cdot 169}{29\cdot 921}$ = ·0564 lb., and thus column 6 in Table 58 has been calculated. The total weight will of course be the sum of the separate weights or ·006639 + ·0564 = ·063039 lb., so that a pound of saturated air will contain $\frac{\cdot 006639}{0564}$ = ·11771 lb. of vapour, and a pound of vapour will saturate $\frac{\cdot 0564}{\cdot 006639}$ = 8·49 lbs. of dry air, and so of the rest as per columns 8, 9, and 10, in Table 58.

CHAPTER VII.

ON EVAPORATION.

(177.) "*Evaporation in Open Air.*"—When a liquid such as water is freely exposed to the atmosphere, the stratum of air in contact with its surface becomes charged with vapour, and thereby becomes lighter than before; thus at 72° a cubic foot of dry air by Table 58 weighs ·0747 lb., but a cubic foot of air saturated with vapour (col. 8) weighs only ·073921 lb., it therefore rises and is replaced by dry air, which in its turn becomes similarly charged and bears off its load of moisture, and so the process continues.

If the air were saturated with moisture to begin with, it could obviously carry no more, and evaporation would cease, but if it were only partially saturated the process would proceed, but more slowly. Thus if we had air at 62° in contact with water at the same temperature, and if the wet-bulb thermometer stood at 52°, showing by Table 60 that the air was half saturated, it is evident that each cubic foot of air could carry off only half the load of moisture, having already half its load to begin with.

(178.) In nature the air is never perfectly dry; in winter it is frequently quite saturated; when it contains about 85 to 90 per cent. of the moisture that would saturate it we consider it damp, 65 per cent. we call dry, 50 per cent. very dry, and even when extremely dry to our feelings it still contains about 35 to 40 per cent. It is matter of common observation that the humidity varies greatly with the season of the year and the hour of the day; the following Table, by Kaemtz, is the result of hourly observation at Halle, and it shows that August is the driest month, air then containing on the average of day and night 66·1 per cent. of the moisture that would saturate it; in the middle of the day when the air is very dry it contains only 49·1 per cent., &c.

	Max.	Min.	Mean.		Max.	Min.	Mean.
January ..	88·9	80·6	85·8	August ..	82·8	49·1	66·1
February ..	86·2	72·5	81·0	September ..	86·1	57·1	72·8
March ..	85·6	67·0	77·3	October ..	87·9	66·1	78·9
April	85·0	56·2	71·3	November ..	89·4	79·6	85·6
May	84·6	53·3	69·2	December ..	88·8	82·6	86·8
June	85·2	55·1	71·0	Year	..	..	76·2
July	84·0	51·2	68·5				

In summer the hour of greatest dryness is between 2 and 3 P.M., and in winter between 1 and 2 P.M. The hour of greatest dampness is in summer between 4 and 5 A.M., and in winter between 6 and 7 A.M. The Table shows, that taking the average of the whole year, day and night, air contains 76·2 per cent. or is rather more than $\frac{3}{4}$ths saturated with moisture.

(179.) "*Effect of Motion in the Air.*"—In perfectly calm air motion ensues only from the change in density produced by the presence of vapour in the air as we have seen (177), the air charged with moisture ascending vertically. If that motion be prevented by a cover, even a loosely fitting one, the confined air soon becomes fully saturated, and not being able to get away with its load, evaporation ceases, and in any case, even if the air be allowed to rise, the motion is very slow, and is retarded by the dry air descending to replace it; but with a wind the stratum of air in contact with the water moves off rapidly horizontally, not waiting to be fully saturated, but taking only a small percentage of that amount. When air moderately dry comes in contact with a surface of water it absorbs moisture at first with avidity, but as it becomes saturated the process proceeds more and more slowly; evidently, therefore, the more rapidly the air is renewed, the more rapidly the evaporation proceeds.

(180.) "*Effect of Temperature.*"—Supposing always that the air and the water have one and the same temperature, the higher that temperature the more rapidly evaporation will proceed, and for two reasons, first, the weight of vapour which a cubic foot of air can carry increases rapidly with the temperature; Table 58 shows (in col. 7) that at 42° air contains ·00044 lb., but at 82°, ·001667 lb., or four times the quantity; and then the difference of density being greater (col. 8) at the higher temperature, the ascensional power is greater and the vapour rises more quickly.

As evaporation takes place only at the surface, it is obvious that with a given volume of water, the greater the surface the more rapidly will any required percentage of it be evaporated.

(181.) Collecting these results, we find that evaporation in the open air proceeds more rapidly:

1st, as the surface exposed is more extended.

2nd, as the temperature of the air, or the water, or both is increased.

3rd, as the dryness of the air is increased.

4th, as the air is most strongly agitated, or moved as wind.

(182.) "*Evaporation at High Temperatures but below Boiling-point.*"—We have so far considered the case in which the water and the air had the same temperature; we will now investigate the phenomena of evaporation with open vessels heated by a fire beneath or otherwise. The experiments of Dalton, made with a vessel 3¼ inches diameter and water at 212°, gave the evaporation at 30 grains per minute in calm air, and 40 grains in a moderate draught such as is obtained over a common fire with an open chimney.

He also found that the amount evaporated at temperatures lower than 212°, was proportional to the force of vapour at the two temperatures; the amount found by experiment is given by the following Table, and we have added from Table 12 the force of vapour for comparison; it will be seen, that the amount

Temperature.	Ratio of Evaporation.	Force of Vapour.
° 212	1·00	29·92
180	$\frac{1}{2}$	15·27
164	$\frac{1}{3}$	10·59
152	$\frac{1}{4}$	7·93
144	$\frac{1}{5}$	6·488
138	$\frac{1}{6}$	5·565

evaporated, is nearly proportional to the force of the vapour at each temperature.

(183.) But strictly, the amount evaporated is proportional to the *difference* of the elastic force of vapour at the given temperature of the water, and of the elastic force of the vapour actually present in the atmosphere. Say we had water and air, both at 32°, and that the air was saturated with vapour; then the tension of the vapour of water and of the vapour in the air being the same, or by Table 12 = ·181 inch of mercury, evaporation would cease. If now the water was heated to 42°, its vapour would have its tension increased to ·267, the air in contact with the water would also be raised to 42°, but the vapour in it being at first still ·181, evaporation would proceed in the ratio ·267 − ·181 = ·086, whereas if the air had been perfectly dry, the ratio would have been ·267 − 0 = ·267, so that only $\frac{\cdot 086}{\cdot 267}$ = ·32 or 32 per cent. of the amount due to dry air, is evaporated. But the effect of the state of saturation of the air becomes very small with high temperatures of water and low temperatures of air: for instance, with water at 212° and air at 32° the elastic force of water is 29·92, and the rate of evaporation would be 29·92 − ·181 = 29·739 with saturated air, and 29·92 with perfectly dry air, the ratio of the two being $\frac{29\cdot 739}{29\cdot 92}$ = ·994, or 99·4 per cent. of the amount due with dry air; so that with great differences of temperature, the effect of humidity in the air is inappreciable, but where the difference is small, it is very influential.

(184.) In Dalton's experiments (182) the temperature of the air is not given, nor its state of humidity, but it will be evident from the above that it is not very important, and the results may be considered as those due to dry air. The evaporation of

30 grains per minute from a vessel 3¼ inches diameter, is equivalent to 520 grains per minute per square foot, or 31200 grains per hour, and admitting that the barometer was at 30 inches, and the air dry, we have the rule $W = 1040 \times D$, in which D is the difference of the elastic force of vapour (in Table 12, &c.) at the temperature of the water, and of the vapour actually present in the air.

(185.) The tension of the vapour present in the air may be found from the indications of the wet and dry bulb hygrometer by Table 60. Thus, say the temperature of the air was 55°, and of the wet-bulb thermometer 50°, or 5° of cold; by Table 60 this would indicate that the amount of vapour in the air was about 70 per cent. of the amount required to saturate it, which by Table 12 is ·433 inch; the tension of the vapour in the air is therefore $\frac{\cdot 433 \times 70}{100} = \cdot 3031$, and if the water had also the temperature of 55°, we should have $1040 \times (\cdot 433 - \cdot 3031) = 135$ grains evaporated per square foot per hour with calm air.

If the water had been at 65°, and the air still at 55°, and containing 70 per cent. of moisture as before, we should have $1040 \times (\cdot 617 - \cdot 3031) = 326$ grains per hour, &c.

If the water had been at 40°, or 15° *colder* than the air, the tension of its vapour would be less than that in the air, and moisture would be *deposited* from the air at the rate of $1040 \times (\cdot 3031 - \cdot 247) = 58$ grains per square foot per hour, &c.

(186.) Let us assume that we have air at 32°, and water at the same temperature, and let us see what happens in evaporating one pound of water at a fixed temperature, say 122°, the ambiant air being dry.

By Table 12, the elastic force of vapour at 122° is 3·621, and that in the air being 0, we have $1040 \times (3 \cdot 621 - 0) = 3766$ grains per hour, as per (col. 2) in Table 59, and from this we obtain also col. 3 and col. 4. The col. 3 shows that ·538 lb. of water will be evaporated per hour, therefore to evaporate one pound, we shall require $\frac{1}{\cdot 538} = 1 \cdot 86$ hour. The heat required to raise the temperature of the water from 32° to 122° and to evaporate it would be by the ordinary rule, $(1178 - 32) = 1146$ units, but by the precise rule in (9) it comes out $1081 \cdot 4 + (\cdot 305 \times 122) = 1119$ units, as per col. 9 in Table 59.

(187.) But there is a loss of heat by radiation from the surface of the heated water, during the 1·86 hour occupied by

TABLE 59.—Of the HEAT REQUIRED to EVAPORATE from 32° 1 lb. of WATER at TEMPERATURES BELOW the BOILING-POINT, from open Vessels exposed to Air at 32°.

Temp.	Water evaporated per Square foot of Surface, per Hour in Calm Air.			Time to evaporate 1 lb. of Water, in Hours.	Heat lost by Radiation from Surface in Units.		Heat carried off by the Air in Units.	The Sum of Latent and Sensible Heat from 32°	Total Heat to evaporate 1 lb. of Water in Units.	Total Heat lost per Hour in Units.	Minimum Volume of Dry Air required to evaporate 1 lb. of Water.	
	Grains.	Pounds.	Depth in Inches.		Per Hour.	Total.					Cubic feet at 32°.	Cubic feet at the Temp. of the Water.
32°	118·24	·0269	·00517	37·1	0·0	000	..	1091°	1091	29	3269	3269
42	277·68	·0397	·00762	25·2	10·7	270	424	1094	1788	71	2208	2252
52	403·52	·0577	·01108	17·3	21·7	375	581	1096	2052	119	1514	1574
62	578·24	·0826	·01588	12·1	33·5	405	605	1100	2110	174	1051	1114
72	816·40	·1166	·02242	8·6	44·9	386	566	1103	2055	239	738	797
82	1135·7	·1622	·03119	6·16	58·1	358	504	1106	1968	319	525	567
92	1561·0	·2230	·04287	4·48	71·2	319	434	1109	1862	415	376	422
102	2119·5	·3028	·0582	3·30	84·7	280	366	1112	1758	533	272	311
112	2842·3	·4060	·0781	2·48	98·9	245	304	1115	1664	671	198	230
122	3766	·5380	·1034	1·86	113·6	211	250	1119	1580	849	144	171
132	4942	·7060	·1357	1·41	128·9	182	202	1122	1506	1068	105	126
142	6414	·9162	·1761	1·09	144·8	158	162	1125	1445	1326	77	92
152	8248	1·1783	·2265	·85	161·5	137	127	1128	1392	1637	55	68
162	10538	1·5055	·2894	·66	179·2	118	97	1131	1346	2039	39	49
172	13262	1·8946	·3642	·53	200·5	106	72	1134	1312	2475	26·8	34·4
182	16609	2·3727	·4561	·42	218·8	92	50	1137	1279	3045	17·4	22·3
192	20630	2·9470	·5671	·34	237·9	81	32	1140	1253	3685	10·1	13·4
202	25431	3·6330	·6984	·275	259·2	71	14	1143	1228	4465	4·4	5·9
212	31200	4·4710	·8563	·224	281·3	63	0	1146	1209	5397	0·0	0·0
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)

the process. The value of R for water by Table 71 is 1·0853, and by the simple rule (267) the loss would be 1·0853 × (122 – 32) = 97·677 units, but applying the correction of Table 80, as explained in (299), we find the *ratio* with 122 – 32 = 90° excess to be 1·163, and the true loss becomes 97·677 × 1·163 = 113·6 units per hour, as in (col. 6) Table 59. The heat lost by radiation during the 1·86 hour is therefore 113·6 × 1·86 = 211 units as per (col. 7).

There is another loss by the air which carries off the vapour, for that air has to be raised from 32° to the temperature of the water, or to 122°. Table 58 shows (col. 10) that each pound of water requires at that temperature 11·65 lbs. of air; the heat thus lost will therefore be 11·65 × (122 – 32) × ·238 = 250 units, as per (col. 8) in Table 59.

(188.) Collecting these results, we have 1119 + 211 + 250 = 1580 units of heat to evaporate 1 lb. of water at 122 (col. 10), or $\frac{1580}{1 \cdot 86} = 849$ units per square foot per hour: this varies with the temperature of the water, as shown by (col. 11). It will also be seen from (col. 10) that the heat required to evaporate a pound of water is not constant, but attains a maximum at about 62°.

Table 59 may be applied to the solution of many practical questions, as we shall proceed to show.

(189.) "*Refrigerators.*"—In the refrigerators commonly used in breweries, &c., a very large surface is necessary, and we can calculate the area for any particular case: say we have to cool 20 barrels, of 36 gallons each, from 202° to 82° in an hour. The work to be done is to dissipate 360 × 20 × 120° = 864000 units per hour. At the commencement, each square foot will lose (col. 11) at the rate of 4465 units, but at the last, 319 units only: taking the loss for each temperature between the extremes, we obtain the average of 1511 units per square foot per hour; so that we require an area of $\frac{864000}{1511} = 570$ square feet. This is the maximum area for calm air; with a very moderate breeze, such as may mostly be reckoned on, the area might be reduced (182) to $\frac{570 \times 30}{40} = 397$ square feet.

There must be free passage allowed for the dry air to enter, and the vapour to depart, &c.; the volume of dry air required is very great; taking a *mean* from 82° to 202° (col. 12), we find that we require 142 cubic feet per pound of water evaporated, the mean quantity evaporated is 1·283 lb. per square foot per hour, in our case 1·283 × 570 = 731 lbs. is evaporated, which will require 731 × 142 = 103802 cubic feet of air.

(190.) "*Evaporating Pans.*"—In concentrating syrups, and in many chemical operations, it is frequently essential that evaporation should be carried on at low temperatures, because high temperatures would injure the product operated on. For such cases the evaporating pan must expose a large area, and must have no cover (179), the depth is unimportant; say we had to evaporate to one-half 10 gallons per hour, that is to say, 5 gallons or 50 lbs. had to be evaporated at a temperature not exceeding 152°, Table 59 shows (col. 3) that at 152° the loss is 1·1783 lb. per square foot per hour; we shall therefore require $\frac{50}{1 \cdot 1783} = 42$ square feet, say 6 feet × 7 feet, or, if circular, about 7 feet 6 inches diameter. The heat required to do the work is by (col. 10) 1392 × 50 = 69600 units, which may be obtained by a fire beneath, consuming $\frac{69600}{6000} = 11 \cdot 6$ lbs. of coal, and a fire-grate about 1 foot square, allowing 6000 units per pound of coal, which is about the economic value of coals for such a case. (See Table 31.)

(191.) "*Evaporation at the Boiling-point.*"—When a liquid is not injured by a high temperature, Table 59 shows that evaporation by boiling is the most economical of all; thus water requires only 1209 units per pound, if the surface is exposed; but if a cover be used, to avoid the loss by radiation, and to which there is no objection as at lower temperatures, the heat is reduced to 1209 − 63 = 1146 units. In this case a large surface is not necessary, the pan may have any convenient form, such as Figs. 59 or 60. Say we take a pan like Fig. 60, with a double or steam case, &c., containing 100 gallons of water, and having the dimensions given by Table 64; the area in *contact with the water* is about 17 square feet, and with 25 lbs. steam, having by Table 11 a temperature of 267°, the difference of temperature between the steam and boiling-water would be 267 − 212 = 55°. Say we require to know the quantity of water at 212°, that would be evaporated in 15 minutes.

By (237) we may admit the evaporative power of a steam-cased vessel to be 330 units for 1° difference in temperature per square foot, and the latent heat of steam at 212° being 966, we shall therefore evaporate $\frac{330 \times 55 \times 17}{966} = 320$ lbs. of water per hour, or $\frac{320}{4} = 80$ lbs., or 8 gallons in 15 minutes.

(192.) We have here supposed that the water was at 212°, or the boiling-point, to begin with: if it had a low temperature, the case would have been different. Say that the water was at 42°, and we require to know what quantity of water would be evaporated from that temperature in an hour. To raise 100 gallons, or 1000 lbs., of water from 42° to 212°, will require $1000 \times (212 - 42) = 170000$ units of heat; the *mean* temperature of the water while being heated will be $\frac{42 + 212}{2} = 127°$, and the difference between the temperature of the steam and the water will be $267 - 127 = 140°$, and with 17 square feet of area the time to raise the water to 212° would be $\frac{170000}{330 \times 140 \times 17} =$ ·216 hour, so that only $1 - {\cdot}216 = {\cdot}784$ hour would be left to do the evaporation work, and the quantity evaporated would be $\frac{330 \times 55 \times 17 \times {\cdot}784}{966} = 250$ lbs., or 25 gallons evaporated to steam. Other forms of apparatus would give different results: see (231), (237), &c.

(193.) "*Condensation Reservoirs to Steam-engines, &c.*"—Say, we take the case of an engine working 24 hours per day, and assume that the temperature of the water for condensation shall be 82°, and the air at 32°. Admitting that a cubic foot of water evaporated is equal to one horse-power, and that the temperature of that water taken from the hot well of the air-pump is 40° higher than the condensation water, or 122°, we shall have $(1178 - 122) \times 62{\cdot}3 = 65789$ units of heat per hour per horse-power to consume or dissipate. The water enters at one end of the reservoir at 122° and departs to the engine at the other end at 82°, and is gradually cooled down 40° in its passage; by (col 11) of Table 59, the rate of loss at 82°, 92°, 102°, 112°, and 122°, are respectively 319, 415, 533, 671, and 849 units per hour, the mean is 557 units, and we shall therefore require $\frac{65789}{557} = 118$ square feet of surface per horse-power for an engine working night and day. The depth in such a case is quite unimportant; but if the engine is to work only 12 hours per day, the surface area might be reduced, because the water would cool during the night, and in that case depth or capacity becomes a necessity, as we will proceed to show.

(194.) Let us assume that 82° shall still be the *mean* temperature of the condensation water; this however will not be uniform, it will be lowest in the morning, after cooling down all night, rising all day till it reaches its maximum at the end of the day's work.

If we admit a variation of 10° each way, we shall have 72° and 92° for the minimum and maximum temperatures of the condensation water. The difference of the temperatures of the water entering from the engine, and departing to it, will still be 40°, for if the engine receives the water 10° colder, it will return it 10° colder also, so that in the morning at starting the water in the reservoir will be 72° at one end, and 112° at the other, and at night, 92° and 132°. The mean loss per square foot by (col. 11) Table 59, from 72° to 112°, is 435 units, and from 92° to 132° it is 707 units; the mean loss during the day is therefore $\frac{435 + 707}{2} = 571$ units. During the night the temperature will be uniform from end to end, being $\frac{92 + 132}{2} = 112°$ at first, and $\frac{72 + 112}{2} = 92°$ at last; the loss per square foot is, by (col. 11), 239, 319, and 415 units, the mean being 324 units per square foot per hour.

The total heat given out by the engine in the 12 working hours, or $65789 \times 12 = 789468$ units, has to be divided into two unequal parts, having the ratio of 571 and 324, hence we have $\frac{789468 \times 571}{571 + 324} = 503600$ units to be dissipated during the day, or $\frac{503600}{12} = 42000$ units per hour, and we shall require $\frac{42000}{571} = 73$ square feet of surface per horse-power.

The question now is, what must be the capacity of the reservoir, or the quantity of water to hold the heat which accumulates during the day, or $789468 - 503600 = 285868$ units. The water being raised 20°, we shall require $\frac{285868}{20} =$ 14293 pounds, or $\frac{14293}{62 \cdot 3} = 232$ cubic feet of water, and as we have an area of 73 square feet, the depth would be $\frac{232}{73}$ 3·2 feet, if the reservoir has vertical sides, &c.; with sloping sides of course the depth would be greater, which is a matter of calculation.

(195.) We find from this investigation that when an engine works day and night the depth is unimportant, and that we require 118 square feet of surface per horse-power; when the engine works only 12 hours per day, 73 square feet will suffice, but in that case the depth must be such as to give 232 cubic feet of water per horse-power.

There will be some advantage in making the reservoir larger than these dimensions where convenient, the water will be cooler in that case and a better vacuum obtained. The following Table gives the particulars of reservoirs in practice, and it shows that reservoirs are frequently made larger than are absolutely necessary for the temperatures, &c., we have assumed.

	Nominal Horse-power.	Surface Area in Square feet.			Capacity in Cubic feet for 12 hours per Day.	
		Actual.	Calculated.		Actual.	Calculated.
			24 hours.	12 hours.		
Brighton ..	100	14464	11800	14600*	87884	46400*
Sutton	40	4500	4720	2900	14742	9280
Ramsgate ..	30	3240	3540	2190	9228	6960
Sevenoaks ..	20	2240	2360	1460	7152	4640

(196.) "*Cold produced by Evaporation.*"—If we cover the bulb of a thermometer with muslin, &c., saturated with water, a considerable amount of cold will be produced, varying with the temperature of the air and the dryness of the atmosphere: see Table 60. Say we take the case of a summer day, with air and water at 72°, and that the wet bulb shows 62°, or 10° of cold. This shows that the film of water is cooled 10°, and the air which carries off the vapour must be reduced to the same temperature as the vapour, and thus it supplies the greater part of the heat required to vapourize the water.

The heat which becomes latent in vapourizing a pound of water at 62° is by the ordinary rule, 1178 − 62 = 1116 units,

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* One engine working 24 hours per day, or two 12 hours.

TABLE 60.—Of the INDICATIONS of the HYGROMETER (Dry and Wet-bulb), from Mr. GLAISHER'S Observations at Greenwich.

Temperature of the Air.	DEGREES OF COLD IN THE WET-BULB THERMOMETER.											
	1	2	3	4	5	6	7	8	9	10	11	12
	DEGREES OF HUMIDITY, SATURATION BEING 100.											
32°	87	75										
42	92	85	78	72	66	60	54	49	44	40	36	33
52	93	86	80	74	69	64	59	54	50	46	42	39
62	94	88	82	77	72	67	62	58	54	50	47	44
72	94	89	84	79	74	69	65	61	57	54	51	48
82	95	90	85	80	76	72	68	64	60	57	54	51

Temperature of the Air.	DEGREES OF COLD IN THE WET-BULB THERMOMETER.										
	13	14	15	16	17	18	19	20	21	22	23
	DEGREES OF HUMIDITY, SATURATION BEING 100.										
32°											
42	30	27									
52	36	33	30	27	25						
62	41	38	35	32	30	28	26				
72	45	42	39	36	34	32	30	28	26	24	23
82	48	45	42	40	38	35	33	31	29	27	26

and as the water itself only yields 10 units, the air must supply the rest, or 1106 units, and the specific heat of air being ·238, we shall require $\frac{1106}{\cdot 238}$ = 4650 lbs., or $\frac{4650}{0747}$ = 62250 cubic feet of air cooled 10° per pound of water. Each cubic foot bears off only $\frac{1}{62250}$ = ·000016 of a pound of water, or about $\frac{1}{1000}$th part of the vapour required to saturate it, and hence it may happen that air after having passed over a large surface of water is only slightly charged with moisture. The quantity of air thus required is so great, that except with a wind, the process of evaporation proceeds very slowly; the observations of Mr. Miller give on an average of six years a yearly evaporation of 30 inches and a maximum of 4·5 inches in a month, as per Table 61.

TABLE 61.—Of the EVAPORATION OF WATER AT NATURAL TEMPERATURES, from observations at Whitehaven by Mr. MILLER.

	Inches.		Inches.
January	·880	August	3·398
February	1·042	September	3·174
March	1·770	October	1·930
April	2·535	November	1·322
May	4·146	December	1·087
June	4·547		
July	4·200	Total	30·032

NOTE.—The above is the mean of six years, during which the mean rainfall was 45·255 inches.

(197.) "*Dryness of Air increased by Heat.*"—The capacity of air for carrying vapour increases very rapidly with the temperature, as is shown by Table 58, so that, for instance, if air at 32° saturated with moisture, and holding in suspension ·00379 lb. of vapour per lb. of air, be suddenly heated to 42°, it will no longer be saturated, because at this latter temperature it could hold ·00561 lb.; it has therefore only $\frac{\cdot 00379}{\cdot 00561} = \cdot 68$, or 68 per cent. of the vapour that would saturate it, and would show by Table 60, about 37°, or 42 – 37 = 5° of cold on the wet-bulb thermometer.

The whole philosophy of drying and evaporation by heated air turns on this fact; in our dwellings too the damp air entering them is not only warmed, but is at the same time converted from damp to dry air, and our sense of comfort arises from both causes.

If air at 32° and saturated with moisture be successively raised as per Table 62 to 72°, we should obtain the given indications of the hygrometer, and the corresponding dryness; but to obtain this result the air must be prevented from absorbing

TABLE 62.

Temp. of Air.	Temp. of Wet Bulb.	Cold.	Degree of Saturation.	
° 32	° 32	° 0	1·00	Saturated.
42	37	5	·68	Dry.
52	42	10	·46	Very dry.
62	46	16	·32	
72	49	23	·23	Intensely dry.

moisture from surrounding objects, which it would do with great avidity.

(198.) With open fires, such as are commonly used in this country, we seldom get an objectionable degree of dryness, the *air itself* is seldom raised to a high temperature, because the only useful heat given out by an open fire is radiant heat, which passes through the air without raising its temperature (268) (298), and is absorbed by the walls, which afterwards heat the air moderately.

Besides, the quantity of air drawn in by the open throat of the chimney is so great, that the dryness is kept down by the large volume of damp air passing through the room. But with close stoves, the air becomes much more highly heated, and the quantity used is diminished, and a very objectionable amount of dryness would ensue, if means were not taken to prevent it. This is usually done by placing a vessel of water on the stove, where it becomes heated, and gives out vapour copiously.

(199.) "*Evaporating Vessels for Stoves, &c.*"—Say we take the case of a stove in which 40 lbs. of coals are burnt per 10 hours, and that we have a vessel half a square foot in area placed on it, and maintained at 122°, while the air in the room was 62°, and external air at 32°: 4 lbs. of coals per hour will require $4 \times 300 = 1200$ cubic feet, or 91·3 lbs. of air, which at 32° will contain by Table 58, $\cdot 00379 \times 91 \cdot 3 = \cdot 346$ lb. of vapour. But during the hour, the small evaporating vessel will add by Table 59 (col. 3), $\cdot 538 \times \cdot 5 = \cdot 269$ lb. of vapour, and we shall then have $\cdot 346 + \cdot 269 = \cdot 615$ lb. of vapour, whereas 91·3 lbs. of air at 62° would require by Table 58 (col. 9), $\cdot 01179 \times 91 \cdot 3 = 1 \cdot 0764$ lb. to saturate it; it therefore holds $\frac{\cdot 615}{1 \cdot 0764} = \cdot 57$ or 57 per cent. instead of ·32 only as per Table 62, which last would be extremely and very uncomfortably dry.

With a vessel of double area, we should have had ·538 lbs. of vapour, and the air would then have contained $\frac{\cdot 346 + \cdot 538}{1 \cdot 0764} = \cdot 82$ or 82 per cent., which would have been too damp, &c. The same result might have been obtained by placing the small vessel in a position where it would have been more highly heated, until its evaporating power was doubled, which by Table 59 would have been at about 152°.

With an open fire, each pound of coal consumes and passes unconsumed up the open chimney, about 6 times the amount of air that coal burnt in a stove requires, and the dryness is thereby reduced.

(200.) "*Evaporation by an Artificial Current of Air.*"—We have seen that a draught or wind causes a more rapid evaporation, the reason for which is explained in (179), &c.; Dalton supposes, that in a very strong current, the rate of evaporation would be double that in calm air. The movement of the air might be effected by a fan, or other mechanical means, or the air used by the fire under the evaporating pan, where that mode of heating is adopted, might be made to pass over the surface before it enters the fire. Taking the case of evaporation at 142°, Table 59 shows (col. 12) that at that temperature, 77 cubic feet of air and 1445 units of heat are required, and admitting that a pound of coals requires 300 cubic feet of air, we shall require for the evaporation of a pound of water at 142° about $\frac{1445}{6000} = \cdot 241$ lb. of coal, and $\cdot 241 \times 300 = 72$ cubic feet of air, so that in this case the air required for combustion and for evaporation, is nearly the same. But at lower temperatures, Table 59 (col. 12) shows that a much greater volume of air is necessary; for instance, at 92° about 5 times as much as at 142°, while the quantity of heat and of coals varies within very narrow limits; so that for temperatures above 152° this plan would answer, but not for lower temperatures. If we allow that the air in passing over the surface of the water should be only *half*-saturated, 162° would be the limit, for at that temperature by (col. 12), 39 feet of saturated, or $39 \times 2 = 78$ feet of half-saturated air, would suffice for the evaporation, being the same as the amount for combustion, which we found to be 72 cubic feet.

CHAPTER VIII.

DISTILLATION.

(201.) It will be seen by Table 9, that the boiling-point of liquids differs considerably, for instance, water boils at 212°, and alcohol at 173°; if a mixture of these two fluids is heated to 173°, the alcohol in it would rise in vapour, leaving the water behind, and thus in theory the whole of the alcohol might be separated, and the whole of the water remain. But practically, it is found that part of the water becomes *entangled* with the alcoholic vapour, and passes away with it; by repeating the process, however, the alcohol may be obtained almost pure.

(202.) The vapour thus evaporated is conducted into a refrigerator, where it is condensed and restored to the liquid state again. The distilling apparatus consists therefore essentially of a closed evaporating vessel, and a condensing vessel, connected together by a pipe to conduct the vapour. The evaporator might be heated direct by the fire, or by steam-pipes circulating through the liquid to be evaporated, &c.

(203.) The quantity of heat necessary to evaporate a given quantity of alcohol may be easily calculated; say we required the quantity of coals necessary to evaporate 30 gallons of alcohol from 62°. A gallon of water weighs 10 lbs., and by Table 46 we see that the specific gravity of alcohol is ·813, therefore in our case the weight of alcohol is $30 \times 10 \times \cdot 813 = 244$ lbs., which will require by Table 1, $244 \times \cdot 622 \times (173^\circ - 62^\circ) = 4646$ units of heat to raise it from 62° to 173°, the boiling-point; to evaporate it we shall require by Table 8, $244 \times 457 = 111508$ units more, or altogether, $4646 + 111508 = 116154$ units, and taking the economic value of a pound of coals at 6000 units, we shall require $\frac{116154}{6000} = 19 \cdot 4$ lbs. of coal to do the whole work; and if it had to be done in an hour, the fire-grate should have an area (120) of about 1½ square foot, and the work being about 2-horse power with that consumption of fuel, we should require (113) about 20 feet effective heating surface in the boiler, or body of the still exposed to the fire, &c. This calculation is for the distillation of alcohol, when it is mixed with foreign bodies, from which we desire to separate it.

(204.) In this country, alcohol is commonly distilled from wort made by the infusion of barley, it contains about 10 per cent. of alcohol; by the evaporation of this to one-half, nearly the whole of the spirit passes over, the liquor thus obtained being composed of ⅕th spirit and ⅘ths water. Say we had a still containing 100 gallons of wort, we should obtain by the first distillation 50 gallons of liquor, composed of 10 gallons of alcohol and 40 gallons of water. The heat required and the size of the apparatus may be easily calculated, the work to be done being composed of three portions, namely, to raise 10 gallons of alcohol from 60° and evaporate it; to raise 40 gallons of water which passes over with it, from 60° and evaporate it,—and to raise the 50 gallons which remain in the still at the end of the operation from 60° to 212°. We will assume that this work has to be done in an hour.

(205.) The alcohol will weigh $10 \times 10 \times \cdot 813 = 81 \cdot 3$ lbs., and the sum of the latent and sensible heats being always the same (8), a pound of alcohol at 60° will require $(173 - 60) \times \cdot 622 + 457 = 527$ units, and in our case we have $527 \times 81 \cdot 3 = 42845$ units for the alcohol. A pound of water requires $1178 - 60 = 1118$ units, or in our case, $1118 \times 400 = 447200$ units; and the water remaining in the still $(212 - 60) \times 500 = 76000$ units, giving a total of $42845 + 447200 + 76000 = 566045$ units. We may fix the size of the boiler, &c., by analogy, with a steam-boiler doing the same amount of work. By (48), we see that about 70000 units are required per horse-power, the work in our case is therefore $\frac{566045}{70000} = 8$-horse power, we must therefore have by (113) and Table 38, say $16 \times 8 = 128$ square feet of *effective* fire-surface; the fire-grate by (120) must be $\frac{128}{18} = 7 \cdot 1$ square feet, and the consumption of fuel will be $7 \cdot 1 \times 14 = 99 \cdot 4$ lbs. of coals.

(206.) The fuel may be economized by heating the wort beforehand by the heat given out in the refrigerator, where the alcoholic vapours are condensed. Assuming the boiling temperature at 200°, to raise the 10 gallons of alcohol from 60° will require $(200 - 60) \times \cdot 622 \times 81 \cdot 3 = 7080$ units, and to heat 90 gallons of water requires $(200 - 60) \times 900 = 126000$ units, altogether $7080 + 126000 = 133080$ units, which is equal by proportion to $\frac{133080 \times 99 \cdot 4}{566045} = 24$ lbs. of coals nearly, the consumption being thereby reduced to $99 \cdot 4 - 24 = 75 \cdot 4$ lbs.

(207.) "*Condensing Apparatus.*"—It is obvious that the apparatus we have described in (230), &c., for heating liquids by vapour, may be applied equally for the condensation of vapour in the process of distillation, and the sizes necessary in any particular case may be determined by the same rules.

The most common form of condenser is the worm in a tub of cold water, as in Fig. 61 and (232); taking the case (205) we may determine the length and diameter of the helix, and the quantity of cold water necessary to effect the condensation. The temperature of the water used will be about the same as that of the atmosphere, taking the hottest month we find, by Table 63,

TABLE 63.—Of the MEAN TEMPERATURE of every Tenth Day in the Year in the Central District of England.

	1st.	11th.	21st.		1st.	11th.	21st.
January	36°·5	35°·6	37°·1	July	61°·2	61°·5	62°·0
February ..	37·2	37·5	38·5	August ..	62·5	61·7	60·6
March	40·1	41·0	41·9	September..	58·8	57·4	55·5
April	43·6	45·0	47·0	October ..	53·5	51·4	49·0
May	50·0	51·3	53·8	November..	46·4	44·0	42·0
June	56·4	57·5	59·8	December ..	41·7	40·2	38·4

the mean temperature of July and August to be say 62°; it may depart at 102°, its *mean* temperature therefore is 82°, and that of the vapour being say 202°, the difference is 202° − 82° = 120°. Admitting 320 units per square foot of surface per hour for 1° difference as in (232), we have in our case 320 × 120 = 38400 units for 120°, and shall require $\frac{42845 + 447200}{38400} = 12 \cdot 7$ square feet of surface; if the pipe is 1¼ diameter outside, its circumference is ·325 foot, and the length must be $\frac{12 \cdot 7}{\cdot 325} = 40$ feet, and the diameter of the helix being say 2 feet or 6·3 feet circumference, we should have $\frac{40}{6 \cdot 3} = 6 \cdot 35$ coils.

(208.) If the temperature of the water in the tub was the same throughout the mass, namely, 82°, we should evidently require $\frac{42845 + 447200}{(82° - 62°) \times 60} = 408$ lbs., or 40·8 gallons of cold water per minute. But the water would not naturally have a uniform temperature, indeed it would require continued stirring to make it so, the water at the bottom (where it should enter) will be nearly cold or 62°, in the middle it will be 82° as we have assumed, and therefore at the top it must be about 102, and if the outlet pipe be taken from that part, as it ought to be, we should only require $\frac{42845 + 447200}{(102 - 62) \times 60} = 204$ lbs., or 20·4 gallons per minute. There is a further advantage too by this arrangement, the alcohol departing from the lowest coil, is cooled down almost to atmospheric temperature, and accidental losses of alcoholic vapour are reduced in amount.

CHAPTER IX.

ON DRYING.

(209.) "*Drying in Open Air.*"—This is the commonest and cheapest mode of all, but in our climate it is too uncertain to be depended on for many manufacturing purposes; it is frequently used however in drying paper, glue, whiting, &c. In such cases a covered building is used to keep off the rain, the sides being open to admit the air freely, but shutters are provided to keep out the air as much as possible on damp days. The rate of drying varies exceedingly, see (177) to (181); generally the air is only about half-saturated with humidity, and drying proceeds rapidly, especially with a wind; but in winter it is often completely saturated, and the drying process is completely suspended. The laws by which drying is governed are explained in the chapter on evaporation.

(210.) "*Drying by Heated Air.*"—The capacity of air for moisture increases rapidly with the temperature, as we have shown in (197), and the efficacy of hot-air drying depends on that fact. The philosophy of the process will be best understood by an illustration.

Let Fig. 67 be a drying-closet, in which the air entering at the bottom becomes highly heated by contact with the steam-pipes, and rising through the closet, finally escapes by the chimney at the top. Say we had 10 lbs. of water to evaporate from 42°; that the external air of a November day was 42° (see Table 63) and *completely saturated* with moisture, the exit temperature being say 102°, and let us admit for illustration that the air at exit is only *half*-saturated with moisture.

Now every pound of air at entry and at 42°, is by col. 9 of Table 58 charged with ·00561 lb. of water, and at its exit at 102° with ·04547 lb. if saturated, but in our case $\frac{\cdot 04547}{2} =$ ·02273 lb. only; it therefore takes up in its passage ·02273 − ·00561 = ·01712 lb. of water, and to carry off 10 lbs. we shall require $\frac{10}{\cdot 01712} = 584$ lbs. of air.

(211.) To evaporate 10 lbs. of water from 42° requires by (8) $1178 - 42 \times 10 = 11360$ units of heat, or the amount that would raise 11360 lbs. of water 1°, and the specific heat of air

(3) being ·238, this is equal to $\frac{11360}{\cdot 238} = 47731$ lbs. of air 1°. But we have only 584 lbs. of air to do the work required, it must therefore be heated $\frac{47731}{584} = 82°$, and the air must enter the closet at $102 + 82 = 184°$.

Thus we have 584 lbs. of air, heated from 42° to 184°, which coming in contact with the wet clothes is cooled down to 102°, the heat thus parted with serving to evaporate the 10 lbs. of water.

The total quantity of heat expended in the process is not only the 11360 units required to evaporate the water, but also that required to raise 584 lbs. of air from 42° to 102°, which is equal to $584 \times (102 - 42) = 8340$ units, making a total of $11360 + 8340 = 19700$ units.

(212.) "*Position of Outlet-opening, &c.*"—In Fig. 67 we have shown the common mode of arranging the inlet and outlet openings, but the plan is a very bad one, as may be seen in the chapter on ventilation (317, &c.); the heated air takes the shortest course to the chimney, and escapes only partially saturated with moisture, and in those parts of the drying-room out of the course of the current the drying process proceeds very slowly. If instead of entering by numerous openings uniformly distributed as in the Figure, the air entered at one place, the case would be still worse.

(213.) The best position for the outlet opening is near the floor, the requisite draught being obtained by a chimney, which may be made of light wood-work. The heated air rises in a body to the roof of the closet as in (320), whence it is gradually drawn down by the action of the chimney; all the horizontal layers of air will have the same temperature throughout, as in Fig 93, and the drying in every part of the room will be equally effective.

(214.) Figs. 68 to 70 give a good arrangement of a drying-closet for linen; the horses are sometimes made of wrought iron, but well-seasoned deal is the lightest, cleanest, and best material in all respects; they are formed with a back and front plate, A B, about 12 inches wide, connected together by horizontal rails, on which the linen is suspended. They are supported by two flanged wheels, running on ⊥ iron rails C, which are prolonged outside the closet, far enough to allow the horse to be drawn out, which is done by the handle D, the upper part being guided by a long wooden rail E E, which is also prolonged outside the closet. The openings, F, are about an inch narrower than A B, and say 2 inches less in height, so that when the horse is in its place, the opening is closed by the plate B, and when drawn out, by the plate A. The air entering from without by the channel G, fills the chamber H, and rises by a series of holes into the chamber J, containing 21 — 9 feet lengths of 4-inch steam-pipes. The floor of the closet is closed all over, except from L to M, which is covered by an open grating of cast iron, the full width of the room, and through this the heated air rises in a body to the roof, where it is distributed, and descends at the back part of the room to the opening N, by which it escapes into the chimney O, and thence into the atmosphere.

(215.) The length of steam-pipes including bends is about 210 feet of 4-inch pipe; if we take the pressure of steam at 10 lbs., its temperature by Table 11 will be about 240°, and at that temperature an *enclosed* pipe as in our case will yield by Table 66, 314 units per foot run, or in our case 314 × 210 = 65940 units per hour, admitting that 10 per cent. is lost by radiation, &c., from the walls of the closet, we shall have 65940 × ·9 = 59346 units available for drying purposes.

(216.) We will assume that the air leaves the steam-pipes and enters the closet at 172° and departs at 82°; it is first heated by the steam-pipes 130°, or from 42° to 172°, and the available heat is sufficient to raise $\frac{59346}{130 \times \cdot 238} = 1914$ lbs. of air to the required temperature.

(217.) By Table 58, a pound of air at 42° saturated with moisture contains ·00561 lb. of water, and at 82, ·02361 lb.; it therefore takes up in passing through the closet ·02361 — ·00561 = ·018 lb. of air, and we have in our case ·018 × 1914 = 34·5 lbs. of water absorbed from the linen per hour. If the wet linen is introduced at 50°, each pound of water in it will require 1178 – 50 = 1128 units of heat to evaporate it, and in our case 1128 × 34·5 = 38923 units per hour. This heat has to be supplied by the air in the act of cooling from 172° to 82°, and we must see that it is capable of doing it; it will yield 1914 × (172 — 82) × ·238 = 41000, or rather more than is necessary.

(218.) If we had assumed 162° for the temperature at entry, we should then have had $\frac{59346}{120 \times \cdot 238} = 2078$ lbs. of air heated which would carry off ·018 × 2078 = 37·4 lbs. of water requiring 37·4 × 1128 = 42187 units of heat. The air in cooling from 162° to 82° would yield 2078 × (162 − 82) × ·238 = 39565 units only, being 42187 − 39565 = 2622 units too little. The conditions assumed in (216) are therefore correct, and the power of the closet may be taken at 34·5 lbs. of water per hour.

(219.) "*Air-chimney, &c.*"—We have in our case 1914 lbs. of air per hour, or ·55 lb. per second, and by Table 19, this is equal at 42° to $\frac{\cdot 55}{\cdot 0791} = 6\cdot 95$ cubic feet at entry from external air, and at 82° to $\frac{\cdot 55}{\cdot 0733} = 7\cdot 5$ cubic feet at exit. But to this last has to be added the 34·5 lbs. of vapour taken up in the closet: this is only $\frac{34\cdot 5}{3600} = \cdot 0096$ lb. per second, or ·0096 × 21·07 = ·2 cubic feet (63) and the volume of air at exit is increased by it to 7·5 + ·2 = 7·7 cubic feet per second.

(220.) The mean temperature of the air in the closet is $\frac{172 + 82}{2} = 127°$, the air in the chimney is at 82°, and both are saturated with moisture. It is necessary to remember this last fact, because the relative weights are affected by the presence of vapour, and the draught-power of the chimney is affected thereby. We assumed in (210) for the sake of varying the illustration, that the air departed only *half*-saturated, but it is a necessary condition where economy is considered that the air should be saturated.

(221.) The condition of the chimney and closet with reference to the creation of a draught current is peculiar, and must be understood before we can calculate the necessary sizes of openings, &c.

Let Fig. 71 be an outline diagram, in which A is the chimney and B the closet both of the same height, the air being at 82° in the chimney, and 127° in the closet. The air in B being lighter than in A in consequence of its higher temperature would ascend, and motion would ensue in the direction shown by the arrows being the reverse of what is required.

Let Fig. 72 be a similar diagram, with a chimney twice the height of the closet, the column in the chimney is now opposed not only by the column in the closet, but also by another one in the imaginary chamber C, which makes up the height of the chimney with air at the external temperature of 42°. The *mean* temperature of the combined column B C is therefore $\frac{42 + 127}{2} = 84^\circ\cdot5$, that in A remaining at 82°; with this height we should therefore still have a reverse draught.

In Fig. 73 we have a chimney three times the height of the closet, and the column in the chimney is now opposed by a column of equal height composed of B, C, D, whose mean temperature will be $\frac{127 + 42 + 42}{3} = 70^\circ$, or 12° colder than the air in the chimney, and we shall now obtain a draught in the right direction.

(222.) We can now calculate the power of the chimney in our case, and will assume for it a height of 28 feet, or four times the height of the closet. The fact that the air at 42°, 82°, and 127°, is all saturated with moisture complicates the case, and it will not be quite correct to take a mean temperature for the columns B, C, D, &c., as we have done with dry air in (221).

The weight of air saturated with vapour is given by (col. 8) of Table 58; for 127° we must interpolate between 122° and 132°, and we have $\frac{\cdot065042 + \cdot063039}{2} = \cdot06404$ lb. per cubic foot, and for the closet 7 feet high $\cdot06404 \times 7 = \cdot44828$, the weight of the column in B; for C, D, &c., we have $\cdot07884 \times 21 = 1\cdot65564$, and together $\cdot44828 + 1\cdot65564 = 2\cdot104$. The air in the chimney at 82° weighs (as a column 1 foot square) $\cdot07226 \times 28 = 2\cdot023$ lbs.

Then by (145), &c., the 28 feet of the lighter air is equal to $\frac{28 \times 2\cdot023}{2\cdot104} = 26\cdot92$ feet of the denser, leaving $28 - 26\cdot92 = 1\cdot08$ feet to produce velocity, which by the laws of falling bodies will give $\sqrt{1\cdot08} \times 8 = 8\cdot3$ feet per second theoretically.

Admitting that the loss of effect by friction, change of direction, and successive enlargements and contractions of the air-passages through the closet (161) which it would be impossible to calculate, this velocity is reduced to half (362) (365), we shall have 4·15 feet per second in the chimney and openings generally; that through the closet would be less, because of the greatly increased area there.

(223.) The chimney O and the opening N having to pass, as we have seen in (219), 7·7 cubic feet of air per second, must have an area of $\frac{7 \cdot 7}{4 \cdot 15} = 1 \cdot 85$ square feet, and may be $1 \cdot 5 \times 1 \cdot 25$ feet; the channel G and the holes in the top of the chamber H must be $\frac{6 \cdot 95}{4 \cdot 15} = 1 \cdot 67$ square feet, &c., &c.

(224.) To facilitate the establishment of the draught when the closet is first heated, it will be well to have an opening P, by which the heated air can pass direct from the steam-pipes into the chimney; when the draught is well established this must be closed, otherwise we should have a waste of heat, and the drying operation would be retarded. A sliding door worked by a rod outside would be the most convenient mode of regulating the size of the opening.

(225.) It will be seen, that with damp air, the chimney must not be less than four times the height of the closet. Where such a height is impracticable from local reasons, a low one may be made to answer by keeping the opening P permanently open, the effect being to increase the temperature of the air in the chimney; but this is an expensive mode of effecting the purpose, and where possible, a high chimney should always be secured.

(226.) In Asylums, Schools, and similar establishments, the size of the drying-closet must be fixed by experience; it will vary considerably with the character of the inmates, &c., Lunatic and Pauper Asylums requiring of course a larger washing and drying apparatus than others. Generally, we may allow 1 square foot of drying-horse for children, say 1½ for men, and 2 for women, estimating the area of the horse by its length multiplied by its height; thus in our case Fig. 68, &c., we have $8 \cdot 5 \times 6 \times 5 = 255$ square feet of surface, which would suffice for say 250 children; or $\frac{255}{1 \cdot 5} = 170$ men; or $\frac{255}{2} = 128$, say 130 women.

In the case of a school for 1200 pauper children of both sexes near London, the work had to be done with thirteen horses, each $9 \cdot 5 \times 6 \cdot 5$ feet, which by our rule would have sufficed for $9 \cdot 5 \times 6 \cdot 5 \times 13 = 802$ children only. It was therefore much too small, and to compensate for that fact, very long hours had to be worked; in damp weather when the whole of the work had to be done by the closet, from 6 A.M. to 9 P.M. for six days per week scarcely sufficed to accomplish it.

An ordinary blanket weighing 3¼ lbs. when dry contains about 6½ lbs. of water when wrung as dry as possible; the closet Fig. 68 would contain five such blankets, and to evaporate the 32·5 lbs. of water in them, we should require $\frac{32\cdot5}{34\cdot5} = \cdot94$ hour, say one hour. A common sheet, weighing 1¾ lbs. dry, contains 2 lbs. water, and 5 such, holding 10 lbs. of water, would be dried in $\frac{10}{34\cdot5} = \cdot29$ hour, or rather more than a quarter of an hour, &c.

CHAPTER X.

ON THE HEATING OF LIQUIDS.

(227.) "*Heating of Liquids by Fire direct.*"—Liquids may be heated directly by a fire, or by steam which may be applied in several ways. When water is heated by a fire the best position for the fire is immediately beneath the vessel, and the worst possible position is the top, for when water is heated it expands, becomes lighter, and ascends, being replaced by colder water which in its turn is heated, and so on until the whole mass is raised at once to the required temperature, but if heat be applied at the top the heated water remains at the top, and may be even boiled there while the water beneath receives no heat whatever, (108) and (311).

(228.) The form of the vessel is unimportant, only it must be such as to receive the heat of the fire readily, Figs. 59 and 60 are common and convenient forms, and for convenience of calculation the dimensions given are for a capacity of 1 gallon, and the dimensions for any other capacity may be found by multiplying by $\sqrt[3]{\text{gall.}}$ required; thus if a copper to hold 125 gallons was required, the $\sqrt[3]{\ }$ of 125 is 5 and the sizes would be given by multiplying all the dimensions in the Fig. 59 or 60 by 5.

(229.) The quantity of water which can be evaporated to steam depends on the area exposed to the action of the fire, and for a vessel of this form we may estimate that each square foot will evaporate 4 lbs. of water at 212° to steam, which is equal to $966 \times 4 = 3864$ units of heat, or to $\frac{3864}{180^\circ \times 10} = 2\cdot14$ gallons, of water at 62° heated 180° (or to 212°) per hour per square foot. The quantity of coals will be about ⅙th of the weight of water evaporated, and the size of the grate may be determined by allowing about 10 lbs. of coal per square foot per hour. Table 64 gives the sizes for coppers up to 300 gallons calculated by these rules.

TABLE 64.— Of the PROPORTIONS of BOILING and EVAPORATING PANS.

Capacity of Copper in Gallons.	Dimensions of Shallow Copper as per Fig. 59.					Dimensions of Deep Copper as per Fig. 60.						Area of Fire-grate in Square feet.	Coals burnt per Hour in lbs.	Pounds of Water at 212° to Steam per Hour.	Gallons of Water heated from 62° to 212° per Hour.
	A	B	C	D	E	F	G	H	J	M	N				
	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.				
20	5½	8¼	1½	15¼	24½	4	14½	1½	20	22	20¼	½	5	29	15·5
50	7½	11½	1½	20½	34	6	20	1½	27½	30	26½	1	10	56	30·0
100	9½	14	1½	25	42	7	25	1½	33½	37	33	1½	15	87	46·5
150	11	16	2	29	48	8	28	2	38	42	38	2	21	124	66·2
200	12	18	2	32	53	9	31	2	42	47	42	2¼	24	144	77·0
250	13	19	2	34	57	9½	33	2	44½	51	45	2½	29	173	92·3
300	14	21	2	37	60	10	36	2	48	54	48	3	31	185	98·8

(230.) "*Heating Liquids by Steam.*"—There are three methods commonly used for heating liquids by steam: by forming a steam-jacket, or double vessel, as in Fig. 60; by a worm circulating through the liquid and filled with steam, as in Fig. 61; and by blowing the steam by a pipe direct into the liquid to be heated.

(231,) Péclet gives an experiment with an apparatus of the form like Fig. 60, in which 1980 lbs. of liquid (beet-root juice) at 39° was raised to 212° in 16 minutes, the steam being at 30 lbs. per square inch (above the atmosphere) and consequently at 274°, and the surface exposed to its action, 25·8 square feet. The work done per hour is $\frac{1980 \times (212° - 39°) \times 60}{16 \times 25{\cdot}8} =$ 50000 units nearly per square foot. The *mean* temperature of the water in this case was $\frac{39° + 212°}{2} = 125°$, and the *difference* of temperature between that and the steam, $274° - 125° = 149°$; and we have therefore from this experiment $\frac{50000}{149°} = 335$ units per square foot per hour for a difference of 1°.

(232.) An experiment was also made with a worm apparatus like Fig. 61, the pipe was 138 feet long and 1·35 inch diameter outside, having a surface of 48 square feet, and was filled with steam at 274°. The vessel contained 880 lbs. of water at 46°, which was heated to 212° in 4 minutes, and in 11 minutes more 550 lbs. of water were evaporated.

(233.) In the first case the work done per hour in heating the water was $\frac{880 \times (212^\circ - 46^\circ) \times 60}{4 \times 48} = 45650$ units per square foot for a mean difference of $274^\circ - \left(\frac{212 + 46^\circ}{2}\right) = 145^\circ$, and for 1° we have $\frac{45650}{145} = 315$ units, agreeing pretty well with the former experiment.

(234.) In the second part of the operation the work done was $\frac{550 \times 966 \times 60}{11 \times 48} = 60400$ units per hour per square foot; the difference between the temperatures was constant, and equal to $274^\circ - 212^\circ = 62^\circ$, therefore for 1° difference we have $\frac{60400}{62} = 974$ units, which is a very high result, and differs unaccountably from (233) deduced from the same apparatus.

(235.) In another experiment with a similar apparatus two worms were used, each 49 feet long, 1·35 inch diameter *inside*, presenting altogether a total exterior surface of 39·5 square feet, filled with 15 lbs. steam having a temperature of 250°, and 132 lbs. of water were evaporated in 5 minutes. The difference of temperature being $250^\circ - 212^\circ = 38^\circ$, we have $\frac{132 \times 966 \times 60}{5 \times 39{\cdot}5 \times 38^\circ} = 1020$ units for 1° difference per square foot per hour; a still higher and incredible result, especially when compared with the following.

(236.) An experiment was made, by Easton and Amos, of London, with a thin welded tube of wrought iron $1\frac{1}{2}$ inch diameter outside and about $\frac{1}{16}$ inch thick, fixed vertically in a vessel of water 12 inches square and 3 ft. 7 in. deep in water as in Fig. 62, steam was turned gently on at A, and the cock B was kept a little open to carry off any water that primed from the boiler, &c.; an air-vent was left open constantly at E, and the water condensed was discharged by an open nozzle at C, and collected in a vessel, D. We can estimate the amount of heat transmitted in two different ways, namely, by the rise in the temperature of the water in F, and by the weight of water condensed, and there should be perfect agreement between the two. The weight of the water in F was 223 lbs., and it was stirred well to produce uniformity of temperature, the surface of the tube in contact with the water was 1·4 square foot. In one experiment the water was raised from 65° to 110°, or 45° in 15 minutes, and 10·218 lbs. of water were collected at D, the quantity at D, calculated from the rise in temperature of the water, should have been $\frac{223 \times 45}{966} = 10 \cdot 38$ lbs., agreeing very nearly with the experimental quantity. The *mean* temperature of the water was $\frac{65 + 110^\circ}{2} = 87^\circ \cdot 5$; and the difference between that and the steam $212^\circ - 87^\circ \cdot 5 = 124^\circ \cdot 5$; and we have therefore $\frac{223 \times 45 \times 60}{12 \times 124 \cdot 5 \times 1 \cdot 4} = 230$ units per square foot per hour, for a difference of 1°. Two other experiments gave 207 and 210 units respectively, and calculating from the water collected, rather less in all three cases. Table 65 gives the collected results.

TABLE 65.—Of the HEATING POWERS of a VERTICAL TUBE with STEAM.

Weight of Water in the Vessel.	Temperature of the Water.				Water collected at D in 15 minutes.		Error.	Units per Square foot per Hour for 1° difference of Internal and External Temperature.
	Min.	Max.	Mean.	Increase.	Actual.	Calculated.		
lbs.	°	°	°	°	lbs.	lbs.	lbs.	
223	65	110	87½	45	10·218	10·38	·082	230
223	60	102½	81¼	42·5	8·67	9·81	1·14	207
223	69	109½	89¼	40·5	8·95	9·35	·4	210

(237.) It is difficult to see a reason for the great discrepancy between these results and those given in (234); a great source of error in such cases is that the air is not expelled, but in the case Fig. 62 the air could escape at *both* ends of the tube, and the difference could not have arisen from that cause. It is almost incredible that in (232) 55 gallons of water could be evaporated in 11 minutes; these results require confirmation, and in the present state of our knowledge we may take the value of a square foot of surface for 1° difference at 230 units per hour for a vertical tube, 330 for a double-bottomed vessel, and say 430 for a horizontal tube or worm. The work done is governed by the *outside* diameter, for with good conductors, such as the metals, the outside of pipes will have sensibly the same temperature as the steam within, see (305) and Table 84.

(238.) The third method of heating fluids with steam, by blowing it direct into the water to be heated, is commonly used for rough purposes, there are some objections to it for refined use; where the steam-boiler is connected with a steam-engine particles of grease may prime over with the steam; the volume of liquid is augmented by the steam condensed to a considerable extent, thus to heat 100 gallons of water from 62° to 212° (or 180°) will require $100 \times 10 \times 180 = 180000$ units of heat, to obtain which we must condense $\frac{180000}{966} = 186$ lbs. of steam, or nearly 19 gallons, which will of course be added to the water to be heated. Another objection is the loud noise with which this method is accompanied, this may be partially obviated by placing the end of the steam-pipe in a small vessel full of small fragments of broken granite, &c., a cover of coarse wire gauze being stretched over the mouth of the vessel to prevent the stone being scattered by the steam.

CHAPTER XI.

ON HEATING AIR.

(239.) "*Heating Air, &c.*"—Air cannot be heated direct by radiant heat for reasons given in (268), but radiant heat can be absorbed by solid bodies, which afterwards give out that heat to the air in contact with them. In our dwellings heated by open fires, radiant heat alone is given out usefully by the fire, the rest being wasted by passing off up the chimney, this radiant heat is absorbed by the walls, and while part of that passes through the walls and is dissipated on external objects, the other part serves to warm the air of the room (298). In heating air therefore the object is to place heated solid bodies which will impart their heat to the surrounding air.

(240.) "*Heating Air by Stoves.*"—Stoves are frequently used in manufactories and for domestic purposes in this country; in the north of Europe they are used exclusively for nearly all purposes.

We may obtain nearly the whole of the heat which any fuel contains by having a long flue or pipe conveying the products of combustion to the outer air, and the length necessary for this purpose may be found from some of Péclet's experiments; he found a considerable difference with different materials for flues.

(241.) "*Sheet-iron Flues.*"—A chimney of sheet iron, 52 feet high, 3½ inches diameter, was mounted on a small furnace, the temperature was taken at the top and bottom, and the velocity of the current was measured directly, by the time which a whiff of smoke (created by a wick soaked in turpentine) occupied in passing from the bottom to the top. The mean of 11 experiments gave the following result: temperature of the air at bottom 536°, and of the top 170°, the exterior air 68°, the velocity of the air 9·8 feet per second, and the total surface of the pipe = 48·6 square feet.

(242.) The mean temperature of the air in the chimney is $\frac{536+170}{2} = 353°$, and the weight of a cubic foot at that temperature by Table 19 is ·049 lb. per foot: the weight of air in the chimney is therefore $\frac{3·5^2 \times ·7854}{144} \times ·049 \times 52 = ·17$ lb., which parts with $536 - 170 = 366°$ of heat, and the specific heat of air being ·238, this is equal to $·17 \times 366 \times ·238 = 14·8$ units of heat; this loss occurs in the time which a particle of air occupies in passing through the pipe, or in $\frac{52}{9·8} = 5·3$ seconds; in an hour, therefore the loss would be $\frac{14·8 \times 3600}{5·3} = 10000$ units, or $\frac{10000}{48·6} = 206$ units per square foot, and the mean temperature of the pipe being $\frac{536 \times 170}{2} = 353°$, the difference between it and the air is $355 - 68 = 285°$, so that for 1° difference we should have $\frac{206}{285} = ·72$ units per square foot per hour.

(243.) For the purpose of experiment the temperatures were low; in practice, the air would probably leave the stove at 1200°, and the end of the chimney at 400°, the mean temperatuure would be 800°, and the excess above the external air, &c., say $800 - 60 = 740°$. With such a high temperature, the loss would be *proportionally*, or per degree, much greater than with the comparatively low experimental temperature, see (299) (301),

&c., and we must apply a correction by Tables 80, 81, and 82. Taking R and A as equal at low temperatures, we have by Table 82, for 353 − 68 = 285° difference, the ratio for R = 2·12 and for A = 1·8, Q therefore = 3·92; but with 800 − 60 = 740° difference, the ratio for R = 7 and for A = 2·2, and Q = 9 2, and hence the loss becomes $\frac{\cdot 72 \times 92}{3 \cdot 92}$ = 1·7 unit for 1°, or in our case 1·67 × 740 = 1236 units per square foot per hour.

Say we take the case of a stove consuming 10 lbs. of coal per hour, each pound of coal will yield a total of about 12000 units (90), of which say ¼th will be dissipated by the body of the stove, thus leaving 9000 × 10 = 90000 units to pass off by the chimney. Each pound of coals requires a minimum quantity of 22·83 lbs. of air (90), but allowing double, say 45 lbs., this air passing into the atmosphere at 400°, or 340° above the temperature of the air in the room, will carry off 45 × 340 × ·238 = 3546 units per lb., or 35460 units for 10 lbs., leaving 90000 − 35460 = 54540 units to be dissipated by the pipe. We shall therefore require $\frac{54540}{1236}$ = 44 square feet of surface in the pipe for 10 lbs. of coals, or 4·4 square feet per pound, with a vertical sheet-iron pipe.

(244.) "*Cast-iron Flues.*"—Similar experiments were made with a cast-iron chimney 54 feet high, 8 inches diameter inside, $\frac{4}{10}$ inch thick, temperature 347° at bottom, 170° at top, and the external air 68°; the velocity was 14·8 feet per second, and the total surface 111·5 square feet. The weight of air in the chimney at the mean temperature 258°, is $\frac{8^2 \times \cdot 7854}{144} \times \cdot 055 \times 54$ = 1·04 lb., which parts with 1·04 × (347 − 170) × ·238 = 43·8 units of heat in $\frac{54}{14 \cdot 8}$ = 3·65 seconds, or $\frac{43 \cdot 8 \times 3600}{3 \cdot 65}$ = 43200 units per hour, or $\frac{43200}{111 \cdot 5}$ = 388 units per square foot of surface, and the mean temperature of the air in the pipe being 258°, the difference between it and the external air is 258 − 68 = 190°, and for 1° we have $\frac{388}{190}$ = 2 units per square foot per hour.

For high temperatures, we may apply the correction as before; with a mean temperature of 258° and 190° difference, the ratio for R by Table 80 is 1·64, and for A by Table 81 = 1·63, Q therefore $= 3 \cdot 27$, whereas with $800 - 60 = 740°$ we found it to be $9 \cdot 2$, hence the loss at the high temperature would be $\frac{2 \times 9 \cdot 2}{3 \cdot 27} = 5 \cdot 6$ units for 1°, and in our case $5 \cdot 6 \times 740 = 4144$ units per square foot per hour.

With a stove as in (243), consuming 10 lbs. of coal per hour, we should require $\frac{54540}{4144} = 13 \cdot 1$ square feet of surface, or $1 \cdot 31$ square feet per pound of coals.

The great difference between cast-iron and sheet-iron flues is very remarkable, a cast-iron flue seems to dissipate $\frac{4144}{1236}$ $= 3 \cdot 35$ times as much heat as a sheet-iron one; it is, however, the clear result of very numerous experiments, and the great loss of the cast-iron flue will be more credible, when we come to calculate the loss of horizontal flues (246).

(245.) "*Earthenware Flues.*"—Péclet made similar experiments on a chimney of earthenware, 43 feet high, $3\frac{1}{4}$ inches inside diameter, $\frac{4}{10}$ inch thick, temperature at bottom 500°, at top 140°, of the external air 68°, velocity $7 \cdot 81$ feet per second, surface $35 \cdot 2$ feet inside, &c. The mean temperature of the air in the chimney was $\frac{500 + 140}{2} = 320°$, and the difference between that and the external air $320 - 68 = 252°$. The weight of air is $\frac{3\frac{1}{4}^2 \times \cdot 7854}{144} \times \cdot 0505 \times 43 = \cdot 1247$ lb.; the loss of heat by this volume is $\cdot 1247 \times (500 - 140) \times \cdot 238 = 10 \cdot 7$ units, in $\frac{43}{7 \cdot 81} = 5 \cdot 5$ seconds, or $\frac{10 \cdot 7 \times 3600}{5 \cdot 5} = 7000$ units per hour for a difference of 252°, or $\frac{7000}{252} = 27 \cdot 7$ units for 1°, or $\frac{27 \cdot 7}{35 \cdot 2} = \cdot 78$ units per square foot of inside surface per hour, being about the same as a sheet-iron flue.

(246.) "*Horizontal Flues.*"—It should be observed, that the foregoing experiments were all made with vertical flues or chimneys, but the loss by horizontal flues would be nearly the same.

Admitting that the outside of a pipe has the same temperature as the inside surface, which is practically true with metal pipes, as shown by (305) and Table 84, we can calculate the loss of heat by horizontal flues heated to 1200° at the end next the stove, and 400° at the other, having thus a mean temperature of 800°, or 800 − 60 = 740° above the external air, by the rules in (302), &c., and we have by Table 82, &c. :—

	R. Ratio.	A. Ratio.		Units per Square foot per Hour.
For 3 inches diam.	$(\cdot 7 \times 7) +$	$(\cdot 6256 \times 2\cdot 2)$	$\times 740 =$	4644
4 "	$(\cdot 7 \times 7) +$	$(\cdot 5745 \times 2\cdot 2)$	$\times 740 =$	4561
6 "	$(\cdot 7 \times 7) +$	$(\cdot 523 \times 2\cdot 2)$	$\times 740 =$	4477
8 "	$(.7 \times 7) +$	$(\cdot 4978 \times 2\cdot 2)$	$\times 740 =$	4440
10 "	$(\cdot 7 \times 7) +$	$(\cdot 4824 \times 2\cdot 2)$	$\times 740 =$	4410

It will be observed, that these numbers are nearly the same as given for an eight-inch vertical pipe of cast-iron (244) by Péclet's experiment, which we found to be 4144 units. But for *sheet-iron* horizontal flues, we must admit the loss to be only 1236 units as in (243).

(247.) "*Effect of Long and Short Flue-pipes.*"—The loss of heat by a stove-pipe per foot run increases from end to end, being of course greatest next the fire where it is hottest. Admitting for illustration, what is not strictly true, that the amount of heat emitted at every point is proportional to the difference of the temperature of the pipe and the external air, let us take the case in which, say $\frac{1}{7}$th of the heat is lost by the first yard of pipe, with a temperature of 1600° to begin with, and external air at 60°; then the air will be reduced to $1600 - \frac{1600 - 60}{7} = 1480^\circ$ at the end of the first yard, and to $1480 - \frac{1480 - 60}{7} = 1277^\circ$ at the end of the second yard, and so on, till at the end of the tenth yard, it is reduced to 382° as shown by Fig. 11, which also gives the loss by each yard, the first losing 220° and the last only 54°, so that the pipe might be shortened considerably without much loss of effect; for instance, the first half of the pipe gives out 64 per cent. of the total heat expended, and the second half only 36 per cent.; in fact, the loss of heat being greater in proportion at high temperatures, the real difference would be much greater than this.

(248.) "*Height of Chimney for Stoves, &c.*"—The velocity of the heated air being known from the volume required to support the combustion and the diameter of the flue, we may easily calculate the height of chimney necessary to obtain that velocity by the rules in (147) and (153).

Thus with the stove we considered in (243), we found that we required 44 square feet of surface in the pipe, and if we assume 6 inches for the diameter, a 6-inch pipe having an area of 1·57 square feet per foot run, we shall require for sheet-iron a length of $\frac{44}{1{\cdot}57} = 28$ feet, or 9·3 yards. Then 450 lbs. of air at 800° by Table 19, is equal to $\frac{450}{{\cdot}0315 \times 60} = 240$ cubic feet per minute, and the area of a 6-inch pipe being ·196 square feet, the velocity will be $\frac{240}{{\cdot}196 \times 60} = 20{\cdot}3$ feet per second, the head for which by Table 47, with a co-efficient of ·93 will be about ·55 lb. per square foot for velocity alone.

The head for friction, by (153) and Table 51, with a 6-inch pipe 9·3 yards long and 240 cubic feet per minute, will be ${\cdot}0000668 \times \left(\frac{240}{100}\right)^2 \times 9{\cdot}3 = {\cdot}00357$ lb. per square inch, or ${\cdot}00357 \times 144 = {\cdot}514$ lb. per square foot, which added to that required for velocity, makes a total of ${\cdot}55 + {\cdot}514 = 1{\cdot}064$ lb. per square foot.

Now a cubic foot of air at 62° weighs by Table 19 ·0761 lb., and at 400° (which is the temperature of the air in the chimney at the end of the flue) ·0461 lb., so that each foot in height gives an unbalanced pressure of ${\cdot}0761 - {\cdot}0461 = {\cdot}03$ lb. per square foot; we shall therefore require $\frac{1{\cdot}064}{{\cdot}03} = 35{\cdot}5$ feet, as the height of the chimney, which is greater than the *length!*

With a slight increase in the diameter of the flue, the height of the chimney would be greatly reduced. Thus a 7-inch pipe having a circumference of $\frac{3{\cdot}14 \times 7}{12} = 1{\cdot}83$ feet, the length would be $\frac{44}{1{\cdot}83} = 24$ feet or 8 yards, and the head for friction is ${\cdot}0000309 \times \left(\frac{240}{100}\right)^2 \times 8 \times 144 = {\cdot}205$ lb. per square foot.

The area of 7 inches being ·267 of a square foot, the velocity with 240 cubic feet is $\frac{240}{{\cdot}267 \times 60} = 15$ feet per second, which by Table 47 is ·31 lb. per square foot. The total is ${\cdot}205 + {\cdot}31 = {\cdot}515$ and the height of chimney $\frac{{\cdot}515}{{\cdot}03} = 17$ feet, instead of 35·5 feet, which we found to be necessary with a 6-inch pipe.

(249.) "*Heating Air by Steam-pipes, &c.*"—We shall see by (302) that the amount of heat given out by heated pipes to the surrounding air per square foot of surface, is not constant, but varies with the diameter, small pipes being most effective, and that 2, 3, 4 and 6-inch pipes yield respectively 327, 303, 291, and 279 units per square foot per hour, when the temperature of the pipe is 210°, and of the external air, &c., 60°. With thin metal such as is commonly used in practice, we may admit that the outside surface has sensibly the same temperature as the steam within, see Table 84 and (305), and assuming $\frac{1}{4}$, $\frac{5}{16}$, $\frac{5}{16}$, and $\frac{3}{8}$ inch for the respective thicknesses, we obtain ·654, ·95, 1·21, and 1·767 as the area or surface of the different pipes per foot run. Taking the 4-inch as an example, we find the loss of heat per foot run to be $291 \times 1{\cdot}21 = 352$ units per hour; the specific heat of air (3) being ·238, this is equal to heating $\frac{352}{\cdot 238} = 1479$ lbs. of air 1°, and as by Table 19 a cubic foot of air at 62° weighs ·0761 lb., we have $\frac{1479}{\cdot 0761} = 19435$ cubic feet of air at 62°, heated 1° per foot run of 4-inch pipe per hour.

Table 66 has been calculated in this way, the loss per square foot for the different temperatures having been calculated as in (302).

"*Pipes enclosed in Narrow Chambers or Channels.*"—When pipes are enclosed in small chambers, whose sides being very near them, become highly heated, they emit very much less heat than when freely exposed to air and distant walls, having a low temperature. This case is investigated in (303) and (382), from which it appears that the loss in the case of an enclosed pipe is about 70 per cent. of the loss when freely exposed; and thus we obtain the numbers in the second part of Table 66.

"*Effect of Variations in Internal and External Temperature.*"—The Table 66 is calculated for pipes freely exposed in a room with air at 62°, but sometimes for drying and other purposes the air is at a much higher temperature, and at other times the temperature is much below 62°. The effect of these changes on the amount of heat emitted is given by Table 67 for a great range of external and internal temperatures; thus say that we required 1000 cubic feet of air per minute, or 60000

TABLE 66.—Of the HEAT emitted by STEAM and HOT-WATER PIPES at different Temperatures, and differently exposed to Cooling Influences.

Mean Temp. of the Pipe.	Temp. of the Air.	Difference.	Units of Heat per Square foot of Surface per Hour.				Units of Heat per Foot run of Pipe per Hour.				Pounds of Air heated 1° per Foot run of Pipe per Hour.				Cubic Feet of Air at 60° heated 1° per Foot run of Pipe per Hour.			
			in. 2	in. 3	in. 4	in. 6	in. 2	in. 3	in. 4	in. 6	in. 2	in. 3	in. 4	in. 6	in. 2	in. 3	in. 4	in. 6
PIPE FREELY EXPOSED TO AIR AND WALLS, BOTH AT ONE TEMPERATURE OF 60°.																		
300°	60°	240°	616	574	553	531	403	545	669	938	1693	2290	2811	3941	22235	28713	36919	51760
280	60	220	543	505	486	467	355	480	587	825	1491	2017	2466	3466	19582	26490	32387	45521
260	60	200	477	443	426	409	312	421	515	723	1311	1769	2164	3042	17218	23233	28421	39952
240	60	180	414	385	370	355	271	366	448	627	1138	1538	1882	2634	14946	20199	24717	34594
220	60	160	356	330	317	314	233	313	384	537	979	1315	1613	2256	12858	17271	21184	29629
210	60	150	327	303	291	279	214	287	352	493	900	1206	1479	2088	11826	15847	19435	27437
200	60	140	299	277	266	256	195	263	322	452	820	1105	1353	1900	10775	14507	17780	24967
190	60	130	271	251	241	231	177	239	292	408	744	1004	1227	1714	9777	13193	16123	22523
180	60	120	245	227	218	209	160	216	264	369	672	907	1109	1550	8830	11920	14573	20368
170	60	110	220	204	196	188	144	194	237	332	605	815	995	1395	7950	10710	13075	18330
160	60	100	196	181	174	167	128	172	210	295	538	722	882	1240	7070	9487	11590	16300
150	60	90	172	159	153	147	112	151	185	260	470	634	777	1092	6176	8331	10210	14350
PIPE ENCLOSED IN A NARROW CHANNEL, WITH BRICK SIDES, &c.: AIR AT 60°; WALLS 158°.																		
300	60	240	431	402	387	372	282	382	468	657	1185	1603	1968	2759	15564	20134	25843	36232
280	60	220	380	354	340	327	248	336	411	578	1043	1412	1726	2426	13707	18543	22671	31865
260	60	200	334	310	298	286	218	295	361	506	918	1238	1515	2129	12053	16263	19895	27966
240	60	180	290	270	259	248	190	256	314	439	794	1077	1317	1844	10462	14139	17302	24216
220	60	160	249	231	222	213	163	219	269	376	685	921	1129	1579	9006	12089	14829	20740
210	60	150	229	212	204	197	150	201	246	348	630	844	1032	1462	8278	11093	13604	19206
200	60	140	209	194	186	179	137	184	225	316	574	774	947	1330	7542	10155	12446	17477
190	60	130	190	176	169	162	124	167	204	286	521	703	859	1200	6844	9235	11286	15766
180	60	120	172	159	153	146	112	151	185	258	470	635	776	1085	6181	8344	10201	14258
170	60	110	154	143	137	132	101	136	166	232	424	571	697	977	5565	7497	9153	12831
160	60	100	137	127	122	117	90	120	147	207	377	505	617	868	4949	6641	8113	11410
150	60	90	120	111	107	103	78	106	130	182	329	444	544	764	4323	5832	7147	10010

TABLE 67.—Of the CUBIC FEET of AIR (at the Temperature of the Room) HEATED from the EXTERNAL TEMPERATURE, per hour per foot run of Steam-pipes, heated to 210°, and exposed to AIR, &c., having the Temperature of the Heated Room.

Temperature of the External Air.	Temperature of Air in the Room.						
	42°	52°	62°	72°	82°	92°	102°
	Cubic feet of Air by 2-inch Pipe.						
12°	451	302	225	186	151	124	102
22	626	403	281	223	176	141	115
32	1253	604	375	279	211	165	131
42	..	1208	513	372	264	198	153
52	..	..	1126	557	352	247	184
	Cubic feet of Air by 3-inch Pipe.						
12	560	406	313	250	204	168	138
22	840	502	391	300	238	192	156
32	1680	813	521	375	285	224	178
42	..	1626	782	500	354	268	208
52	..	..	1563	750	475	335	250
	Cubic feet of Air by 4-inch Pipe.						
12	685	496	382	307	249	206	170
22	1028	661	478	369	291	235	191
32	2056	992	637	461	349	274	219
42	..	1984	955	615	437	329	255
52	..	..	1910	922	583	411	306
	Cubic feet of Air by 6-inch Pipe.						
12	960	702	536	428	351	264	240
22	1440	936	671	514	409	330	270
32	2881	1404	894	643	491	385	309
42	..	2809	1342	857	614	462	360
52	..	..	2683	1285	818	578	432

cubic feet per hour at 82°, the external air being at 32°; then by the Table, we should require $\frac{60000}{211} = 284$ feet of 2-inch pipe, or $\frac{60000}{285} = 210$ feet of 3-inch, or $\frac{60000}{349} = 172$ feet of 4-inch, or $\frac{60000}{491} = 122$ feet of 6-inch pipe, &c., &c.

(250.) "*Air-Cocks, &c.*"—Water at ordinary temperature always contains a considerable quantity of air, which is expelled by heat, and passes off mixed with the steam, retarding the communication of the heat to the pipe containing it. It is therefore necessary to get rid of it, and this can be done most simply by small cocks fixed here and there along the course of the pipes, usually on the top; but air being heavier than steam at the same temperature and pressure as shown by (173), theory indicates that the air-cock should be at the bottom of the pipe, and in one or two cases in which it has been tried, practice does not disprove it, so that the apparatus used to discharge the water condensed may sometimes be used to get rid of the air also.

(251.) "*Apparatus for discharging condensed Water.*"—The quantity of water condensed by steam-pipes is shown by (302); it is very considerable, and being constant, the apparatus should be self-acting. Where the steam is taken from a low-pressure boiler, an inverted syphon is the most simple contrivance for this purpose; but when the heating-pipes are at the ground level, the syphon requires an excavation to be made for it, which may be difficult and sometimes impracticable; for 7 lbs. steam the column of water required is, by Table 41, about 16 feet, and allowing a margin for fluctuation in pressure, the syphon should not be less than 20 feet long for that pressure. There is a considerable loss of effect by the use of this apparatus, arising from the circumstance that the water is discharged at a very high temperature; in fact, at or near that of the steam, so that with steam at 210° and exterior air at 60° we might have $1178 - 60 = 1118$ units of heat per lb. of steam; but if the water is discharged at 210°, we only obtain $1178 - 210 = 968$ units, or $\frac{968}{1118} = \cdot 86$, or 86 per cent. of the available heat, and 14 per cent. is wasted.

(252.) Figs. 51 and 52 show a simple apparatus by which this loss may be avoided, the water being retained till nearly cold; and it has the further advantage of discharging the air as well as the water: A is a $\frac{3}{4}$-inch gas-pipe about 8 or 9 feet long; B B two rods of wrought iron about $\frac{5}{16}$ inch diameter, secured by nuts at the upper end to the winged socket D, which is fixed to the pipe by the pinching-screw E, and similarly secured at the lower end to the cross-bar C, which carries a screw G by which the valve H is adjusted; these rods are guided by passing loosely through wings F F on the valve seat. The best position for this apparatus is the vertical one as shown by Fig. 52; but where a vertical position is inconvenient, it may be fixed at only a slight angle with the horizon. The pipe A, communicating with the bottom of steam-pipe J, will, when filled with steam, have the same temperature as that steam, and the screw G must then be regulated so as just to close the valve. The water that is condensed accumulates in A, and becoming cold, the pipe A is shortened, while the rods B B are not affected; the valve seat therefore retreats from the valve, which opens and allows the cold water to escape; but when hot water or steam follows, the pipe A, becoming heated, is expanded again to its normal length, and the valve is again closed, and so on. The variation in length is not very great, but with the length given, is found by experience to be sufficient for the purpose. Taking the expansion of wrought iron from Table 14, and allowing that the water escapes at 110° while the steam is 210°, the expansion for a length of 108 inches is $\cdot 000006689 \times 108 \times 100 = \cdot 072$, or about $\frac{1}{14}$th of an inch. A brass tube about 6 feet long would expand about the same as an iron one 9 feet long, and may be used where a great length is objectionable.

(253.) Fig. 54 shows a valve box and float which is sometimes used for getting rid of the condensed water. A is a float of copper, &c., attached by a loosely-fitted ring or shackle B to the valve C; a ring of metal D is cast on the bottom, and serves to receive and retain the float in position; a similar ring E on the cover serves to prevent the float rising too high, and thereby drawing the valve out of its seat, &c. There is some difficulty in practice in keeping the float tight, especially with high-pressure steam, and the valve is apt to stick and give trouble. The steam, &c., enters at F, and the condensed water escapes at G.

(254.) It is very desirable that the water should not be wasted, but wherever possible it should be conducted back to the boiler, and used over and over again; ordinary water contains salts of lime, &c., &c., which in the case of a steam-boiler do not pass off with the steam, but are deposited on the internal surface of the boiler, and are very destructive, by obstructing the transmission of the heat and causing the metal to become unduly heated; but by feeding the boiler with the water returned from the steam-pipes, which is of course pure distilled water, this may be avoided. In most cases where there is no steam-engine, a small donkey engine and pump is used; but latterly the Giffard's Injector has been used, and is by far the neatest and cheapest apparatus for this purpose.

(255.) "*Expansion of the Pipes.*"—Arrangements should be made to allow for the expansion caused by the great changes of temperature which steam-pipes experience; the simplest course is to mount them on rollers like Fig. 53, allowing them liberty at the ends to expand freely. In long ranges the expansion is considerable; thus with high-pressure steam at 300° and air at 50°, we have 250° difference between the extremes, and a pipe 100 feet long expands $\cdot000006167 \times 250 \times 100 \times 12 = 1\cdot85$ inch; with low-pressure steam at 210° and air at 50°, the expansion would have been 1·184 inch.

(256.) Where the movement of the pipe is inconvenient, an expansion joint must be used. Fig. 56 shows a good form for this; all the fitting parts should be roughly turned and bored, but should be slack fits; $\frac{1}{16}$th play may be allowed with advantage all over. C is a wrought-iron safety ring riveted on the end of the pipe A, the object of which is to prevent accident by the pressure of the steam blowing the joint apart; this pressure is considerable, for instance, with a 4-inch pipe, $\frac{1}{2}$-inch thick, and 45 lbs. steam, there is a force of 882 lbs. tending to blow the spigot out of its socket; and if by an oversight in fixing, a sufficient resistance is not provided, a serious accident may ensue. This cannot happen with Fig. 56, the wrought-iron ring C being prevented from coming out by a second ring D, which again is kept in position by the packing and gland bolts. Care should be taken in *fixing*, so as to adjust the length of the pipes, that C has room to move in the *right direction*, the effect of expansion being to increase the length; C should be nearly touching D when cold, and the distance from C to E should of course be sufficient to allow for the utmost expected variation in length; if by carelessness C is fixed in contact with E, there can be no expansion, and the apparatus is useless.

(257.) "*Heating Air by Hot-water Pipes.*"—In Fig. 47 let A B C D be a pipe filled with water, and let heat be applied at A; the column A B will be heated and expanded, and becoming thereby lighter than the counterbalancing column C D, will begin to ascend in the direction shown by the arrow. Let us suppose that at A the water is heated to 210°, and that the air is at 60°, the pipe will give out heat to the air, so that the heat imparted to it at A is gradually and progressively lost; and assuming that it returns to A at 110°, and that it loses equal amounts of heat for equal distances from A, we obtain the series of temperatures given in the figure. We can calculate the velocity of the water in any particular case; say that the height

A B was 40 feet, B C and A D 10 feet each, and the pipe 4 inches diameter, then by Table 18 we find the relative specific gravities of water at 190° and 140° to be about as ·9659 to ·9822, and the column C D being 40 feet, will be opposed by the column A B, having a weight equivalent to $\frac{40 \times \cdot 9659}{\cdot 9822} =$ 39·336 feet; the two columns, therefore, do not balance one another, but there is a difference of $40 - 39 \cdot 336 = \cdot 664$ feet, or 8 inches, tending to produce motion in the water, and which by the principles of hydraulics would produce a velocity equal to the discharge of 70 gallons per minute. But 70 gallons, or 700 lbs. of water per minute cooled 100°, would yield $700 \times 100 \times 60 = 4200000$ units per hour, whereas by Table 66 a 4-inch pipe at a mean temperature of 160°, gives 210 units per foot, or in our case $210 \times 100 = 21000$ units per hour only, so that we have an excess of power, and to obtain the conditions shown by Fig. 47, which we assumed for the purpose of illustration, the pipe would have to be throttled by a valve or cock, &c.; or if left free, the water would return to the boiler at A at a temperature very little less than it left it, in which case the pipe, being at a higher temperature throughout, would impart more heat than we found for it.

(258.) Fig. 48 shows a pipe precisely similar to Fig. 47, but here the fire is at B, half-way up the pipe; assuming the same temperatures as before, it will be seen by the figures that the *mean* temperature of both columns is exactly the same, or 160°; of course in such a case there is equilibrium, and no motion will ensue, nor will any forcing of the fire produce it. This will serve to show that in all cases the fire should be as near to the bottom of the ascending column as possible; when quite at the bottom, a very short column indeed will suffice to produce the necessary motion.

(259.) Let Fig. 49 be a pipe 500 feet long, 4 inches diameter; A B and C D each one foot, and let A B have a temperature of 210° and C D 160°, the water returning to B at 110°. At the given temperatures of 210° and 160°, Table 18 gives about ·9585 and ·9761 for the specific gravities of water, so that A B being 12 inches, C D would be equal to $\frac{12 \times \cdot 9585}{\cdot 9761} = 11 \cdot 783$ inches; there would therefore be an unbalanced pressure of $12 - 11 \cdot 783 = \cdot 217$ inch tending to produce motion. By Table 68, which is calculated by Prony's formula, we may find

TABLE 68.—Of the FRICTION of WATER in PIPES.

Velocity in Feet per Second.	$\frac{H}{L} \times d$.	Theoretical Head to produce the Velocity in Column 1.	Velocity in Feet per Second.	$\frac{H}{L} \times d$.	Theoretical Head to produce the Velocity in Column 1.
		inches.			inches.
·01	·000008866	·0000187	·13	·0001943	·003168
·02	·00001870	·0000750	·14	·0002169	·003675
·03	·00002813	·0001687	·15	·0002394	·004118
·04	·00004148	·0003000	·2	·0003702	·007500
·05	·00005437	·0004687	·25	·0005266	·011717
·06	·00006830	·0006750	·3	·0007080	·016875
·07	·00008320	·0009187	·35	·0009154	·022968
·08	·00009920	·001200	·4	·001148	·030000
·09	·0001161	·001518	·45	·001406	·037970
·10	·0001341	·001875	·5	·001700	·046850
·11	·0001532	·002268	·55	·00200	·056720
·12	·0001732	·002700	·6	·00233	·06750

the velocity which that small head would generate, for $\frac{H}{L} \times d$ is in our case $\frac{\cdot 217}{6000} \times 4 = \cdot 000144$, which by the Table is equal to a velocity of about ·105 foot per second, an extremely small velocity, but more than sufficient for the purpose, as we shall see presently. By Table 66 we find that a 4-inch pipe at 160° gives 210 units per hour; we have therefore in our case $210 \times 500 = 105000$ units per foot run per hour, or $\frac{105000}{3600} =$ 29·1 units per second, and each foot of water in a 4-inch pipe weighing 5·4 lbs., losing in our case 100°, this is equal to $5 \cdot 4 \times 100 = 540$ units per foot, and to obtain 29·1 units, the water must move with a velocity of $\frac{29 \cdot 1}{540} = \cdot 053$ foot per second; we found the velocity really obtained to be ·105 feet per second, or double the minimum necessary, so that with the given conditions, vertical columns only one foot high will work a pipe 500 feet long effectively.

(260.) To facilitate these calculations, we have given in Table 69 the head to be obtained with different temperatures of vertical columns of water in the descending pipe, that of the ascending column being 212°. In Table 70 we have given the minimum velocity of current necessary in different sized pipes

100 feet long, calculated as in (259). It will be found that when the temperatures are constant, the velocity *necessary* varies in the simple ratio of the length of pipe, so that the Table applies to all cases. Thus a pipe 1000 feet long will require ten times the velocity for a pipe 100 feet long, &c.

(261.) Table 68 will enable us to calculate the velocity of the current, having the head given by Table 69, &c., &c.

TABLE 69.—Of the HEIGHT of EQUIVALENT COLUMNS of WATER at DIFFERENT TEMPERATURES, the Height at 212° being 12 inches.

Temperature of Water.	Height of Column in Inches.	Difference from 212° in Inches.	Temperature of Water.	Height of Column in Inches.	Difference from 212° in Inches.
212°	12·000	·000	122°	11·647	·353
202	11·954	·046	112	11·611	·389
192	11·908	·092	102	11·599	·401
182	11·868	·132	92	11·570	·430
172	11·824	·174	82	11·550	·450
162	11·783	·217	72	11·532	·468
152	11·746	·254	62	11·518	·482
142	11·710	·290	52	11·508	·492
132	11·677	·323	42	11·502	·498

TABLE 70.—For HOT-WATER PIPES, showing the Minimum Velocity of Current necessary for the renewal of the Heat in a Pipe 100 feet long, with different Temperatures of Pipe, exposed to Air at 60°.

Temperature of the Water in the Pipe.				Diameter of Pipe in Inches inside.			
Mean.	As it leaves the Boiler.	As it returns to the Boiler.	Difference.	2	3	4	6
				Velocity in Feet per Second.			
205°	210°	200°	10°	·3900	·2290	·164	·1040
200	210	190	20	·1993	·1190	·0817	·0510
190	210	170	40	·0904	·0535	·0367	·0230
180	210	150	60	·0545	·0326	·0224	·0139
170	210	130	80	·0368	·0219	·0151	·0094
160	210	110	100	·0262	·0156	·0107	·0067
150	210	90	120	·0191	·0119	·0786	·0049

NOTE.—The velocity necessary is simply proportional to the length of the pipe; thus a pipe 500 feet long requires five times the velocity given by the Table for 100 feet, &c.

1st. Having the length, diameter, and velocity given, to find the *head*, take from Table 68 the number in column 2 opposite to the given velocity; this multiplied by the length of the pipe in inches, and divided by the diameter of the pipe in inches, will give the required head in inches, the height of column to give which must be calculated by Table 69.

2nd. To find the *velocity*, multiply the given head in inches (obtained by Table 69) by the diameter in inches, and divide by the length of the pipe in inches, and find the nearest number thereto in column 2, then opposite to that, in column 1, is the required velocity.

(262.) We found in (259) that columns 1 foot high were sufficient to work a pipe 500 feet long; but to obtain that result we were obliged to allow the water to return to the boiler very much reduced in temperature. The *mean* temperature of the pipe being low, we obtained from it only 210 units per foot, whereas we should have had 352 units if the water had retained throughout the temperature of 210°. It is desirable, therefore, to give long vertical columns where possible, as the temperature may then be higher, and the system more efficient. With short pipes very little head is required, but with long ones so much head is required that it cannot be given by moderate columns of water without allowing great differences in the temperature of the exit and return water; so that in general long pipes are not so efficient in proportion as short ones. The velocity necessary for the renewal of the heat being proportional to the length, and the head or height of vertical columns proportional to the velocity squared, multiplied by the length, it follows that the height of column should be proportional to the cube of the length of the pipes, so that for lengths 1, 2, 3, the column should be in the ratio 1, 8, 27.

(263.) Let Fig. 50 be a 3-inch pipe 400 feet long, and say that we require to find the height of the columns A and B for different temperatures:—say A is 210° and B 190°, the mean temperature is then 200°. Table 70 gives ·119 as the necessary velocity for a pipe 100 feet long. In our case we require $\cdot 119 \times 4 = \cdot 476$ feet per second, which is between ·45 and ·5 in Table 68; we may take $\frac{H}{L} \times d$ in column 2 therefore at ·00155, which (261) multiplied by the length and divided by the diameter, both in inches, or in our case $\frac{\cdot 00155 \times 4800}{3}$

= 2·48 inches head. By Table 69, for 190° we have a head of ·092 inch for a column 1 foot high, therefore we shall require $\frac{2\cdot048}{\cdot092}$ = 27 feet columns in our case; and the pipe will yield by Table 66, 263 units per foot run. Again, say A is 210° and B 170°, the mean temperature being 190°. Table 70 gives the velocity at $\cdot0535 \times 4 = \cdot214$ foot per second, which may be taken at ·0004 in column 2 of Table 68; and the head is $\frac{\cdot0004 \times 4800}{3} = \cdot64$ inch, which again by Table 69 is equal to $\frac{\cdot64}{\cdot174} = 3\cdot7$ feet columns; the pipe will yield in this case for a mean temperature of 190°, 239 units per foot instead of 263 as we found it before.

(264.) We have seen that it is desirable to place the fire as near the bottom of the ascending column as possible, and in (258) we have given a case that is impracticable where the fire was midway in the height of that column. In some cases the excavation necessary for placing the fire at the bottom is very objectionable; and as it is *possible* to work a system of pipes with the fire considerably *above* them, by using lofty vertical columns, we will investigate the case, without, however, recommending its adoption where it can possibly be avoided.

(265.) Let Fig. 55 be a pipe 4 inches diameter, and altogether 200 feet long, of which the two columns A and B are each 40 feet long, and the pipes D, E, F, and G laying at the same level in the room to be heated, and let the fire be at C, 5 feet above the general level of the pipes in the room. If we allow that the water leaves the fire at 210° and returns at 190°, losing 20° in 200 feet, it follows that 1° is lost by each 10 feet in length; at H therefore it will be 189°·5; at J $210 - \frac{35}{10} =$ 206°·5; and at K $206\cdot5 - \frac{40}{10} =$ 202°·5. The mean temperature from H to C will therefore be $\frac{189\cdot5 + 190}{2} =$ 189°·75, from C to J $= \frac{210 + 206\cdot5}{2} =$ 208°·25, and of the whole column A $\frac{(189\cdot75 \times 5) + (208\cdot25 \times 35)}{40} =$ 205°·94, say 206°. The mean temperature of the column B is $\frac{206\cdot5 + 202\cdot5}{2} = 204\cdot5$, so that it is 1°·5 colder than A; and this is all the motive-power at our disposal.

By Table 18 the expansion of water between 202° and 212° is ·000411 for 1°, and a column of water 40 feet or 480 inches long, heated 1°·5, will expand $\cdot 000411 \times 1\cdot 5 \times 480 = \cdot 296$ inch, which is the head to produce motion. By Table 68, $\frac{H}{L} \times d$ is in our case $\frac{\cdot 296}{200 \times 12} \times 4 = \cdot 000493$, which by col. 2 may be estimated at ·24 feet velocity per second. The velocity necessary for the renewal of the heat in the water for a 4-inch pipe, whose mean temperature is 200°, we shall find by Table 70 to be ·0817 foot for a pipe 100 feet long, in our case therefore we shall require $\cdot 0817 \times 2 = \cdot 1634$ foot per second, so that the velocity of ·24 foot actually obtained is sufficient.

CHAPTER XII.

ON THE TRANSMISSION OF HEAT, AND LAWS OF COOLING.

(266.) The cooling of heated bodies may be effected by three methods:—1st, by radiation; 2nd, by contact of cold air; and 3rd, by conduction. Putting U for the total loss from all causes, and R′, A′, and C′ for the respective losses by radiation, contact of air, and conduction, we have $U = R' + A' + C'$.

"*Loss of Heat by Radiation.*"—Let P, Fig. 74, be a plate say of building-stone, 1 foot square and 1 inch thick, having both its surfaces, S, S′, as also the air in contact with S, at a constant temperature of 60°; and let the surface S be exposed to distant walls, W W W, maintained at 59°, or 1° lower than S. Under these conditions, the surface S will obviously lose no heat by contact of air, because that air has the same temperature as itself; for a similar reason, there will be no loss by conduction, because S′ has the same temperature as S; but S will send out in all directions towards W W, rays of radiant heat, which will proceed in straight lines through the air, until they are intercepted and absorbed by the walls. The amount of heat thus lost will vary exceedingly with the nature of the surface. This is shown by Table 71, which gives the loss in units of heat per square foot per hour for 1° difference between S and W, as in our case; and for building-stone this loss may be taken at ·736 unit per hour, &c. If W had the same temperature as S, radiation would cease altogether; if, on the other hand, the temperatures were reversed, S being in that case 1° *lower* than W, W, it would absorb the same amount of heat as it emitted in the former case; the radiant and absorbing power of bodies being equal to one another.

TABLE 71.—Of the RADIATING and ABSORBING POWER of BODIES, being the Units of Heat emitted or absorbed per square foot per hour for a Difference in Temperature of 1° Fahr. From the experiments of PÉCLET.

	Value of R.
Silver,* polished	·02657
Copper ,,	·03270
Tin ,,	·04395
Zinc and Brass, polished	·04906
Tinned Iron ,,	·08585
Iron, Sheet ,,	·09200
Lead ,,	·13286
Iron ,, ordinary	·56620
Glass	·59480
Cast Iron, new	·64800
Chalk	·6786
Cast and Sheet Iron, rusted	·6868
Wood Saw-dust, fine	·7215
Building Stone, Plaster, Wood, Brick	·7358
Sand, fine	·7400
Calico	·7461
Woollen Stuffs, any Colour	·7522
Silk Stuffs, Oil Paint	·7583
Paper, any Colour	·7706
Lamp-black	·8196
Water	1·0853
Oil	1·4800

(267.) For ordinary atmospheric temperatures of absorbing surfaces, say 50° to 60°, and small differences between S and W, say 30°, we may admit that the loss of heat is simply proportional to that difference; and that $R' = R \times (t - T)$, in which R' is the loss by radiation, in units per square foot per hour, R = the radiant power of the body from Table 71, t = the temperature of the radiant, and T = the temperature of the absorbant; but for high temperatures of T, and great differences between t and T, the loss of heat is much greater, following a complicated law, for which Dulong has given a rule that agrees well with experiment, up to a very high temperature. (See (299), Chapter XIII.)

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* Copper silvered.

(268.) Radiant heat has the remarkable property of passing through *moderate* thicknesses of air or gas, without suffering appreciable loss or heating the air sensibly. With very great thicknesses, however, such as several miles, the loss of heat becomes evident. This is shown by the diminished power of the sun at and near its rising and setting, when its rays passing very obliquely through the atmosphere, traverse a much greater thickness of air than at mid-day. The variation from this cause is given by Table 72 from the observations of Mr. Daniell. For the same reasons, the solar radiation varies throughout the year with the varying altitude of the sun. This is shown by Table 73. For ordinary cases we may admit that air and gases cannot be heated directly by radiant heat, but only by contact with heated bodies.

TABLE 72.—Of the VARIATION of SOLAR RADIATION, throughout the day in the month of June. Mr. DANIELL.

Time.	Temperature.		Difference.	Time.	Temperature.		Difference.
	In Sun.	In Shade.			In Sun.	In Shade.	
	°	°	°		°	°	°
9 A.M.	93	68	25	2 P.M.	143	75½	67½
9.30	103	69	34	2.30	138	76	62
10	111	70½	40½	3	138	76½	61½
10.30	119	71	48	3.30	132	77	55
11	124	71½	52½	4	124	76	48
11.30	125	72½	52½	4.30	123	77	46
12	129	73	56	5	112	76	36
12.30	132	74	58	5.30	106	75	31
1	141	74½	66½	6	100	73	27
1.30	140	75	65				

(269.) The loss by radiation is not affected by the *form* of the radiant body, a cube, a sphere, a cylinder, &c., &c., will radiate the same amount of heat with equal areas under the same conditions, so long as the body is not of such a form as to *radiate to and from itself*. The colour of the surface seems to have no effect on the radiant power of bodies, at least this is true of paper and woven fabrics.

Table 73.—Of the Variation of Solar Radiation throughout the Year. Mr. Daniell.

	Mean Maximum Temperature of the Air.	Mean Maximum Solar Radiation.	Maximum force of Solar Radiation.
	°	°	°
January..	39·6	4·4	12
February	42·4	10·1	36
March	50·1	16·0	49
April	57·7	28·1	47
May	62·9	30·5	57
June	69·4	39·9	65
July	69·2	25·8	55
August	70·1	33·1	59
September	65·6	32·7	54
October	55·7	27·5	43
November	47·5	6·7	24
December	43·2	5·4	12

The loss by radiation is not affected by the *distance* of the absorbant, thus in Fig. 74, if the space traversed were a vacuum, it would be quite immaterial, whether W W were inches, or millions of miles, distant from S, the rays of heat would pass on till they met with the cold body which had attracted them. Thus the heat we receive from the sun is radiant heat, which has travelled 95,000,000 of miles through space. The absorbant of radiant heat need not be a *solid* body, celestial space has a low temperature, and although it is a vacuum, or nearly so, yet it attracts the radiant heat due to its temperature. Radiant heat seems to be of two kinds, one luminous, such as that of the sun, a candle, &c., and the other obscure or dark rays, such as are given out by a vessel of hot water.

(270.) "*Loss of Heat by Contact of Air.*"—The loss of heat by contact of cold air, is independent of the nature of the surface, so that cast-iron, stone, wood, &c., &c., would lose the same amount of heat, under the same conditions of temperature, but the *form* of the body affects the result considerably, so that a plane, a sphere, and a cylinder will lose different amounts of heat per square foot in the same time.

Let P, in Fig. 75, be a vertical plane of *any* material, 1 foot square, having its surface S maintained at 60°, and let S′ and W W W have also the temperature of 60°, while the air in contact with S is at 59°. There will in this case be no loss of heat from S by radiation or conduction, because W and S′ are of the same temperature as S, but heat will be given out to the cold air; and experiment has shown that the amount for 1°, as in our case, is ·5945 unit per square foot per hour for a plane 1 foot high, but it will not be the same per square foot for any other height, for reasons that are given in (272).

(271.) For small differences of temperature (between the air and the body), say up to 30°, we may admit that the loss of heat is simply proportional to that difference (see Table 81), and the rule becomes $A' = A \times d$, in which d is the difference of temperature between the air and the body; A' = the loss of heat by contact of air, in units per square foot per hour; and A = the loss for 1° difference of temperature between the body, and the air in contact with it. This last will have a value, varying with the *form* of the body, as we shall see presently.

With great differences of temperature, Dulong has shown that the loss of heat increases in a much higher ratio than that difference, so that, for instance, when a body is 450° above the temperature of the air, the loss per degree is double the loss with small differences of 20 or 30 degrees (see (301) and Table 81).

(272.) If we investigate the loss of heat by a vertical plane, we shall be able to see the reason for the varying value of A. A vertical plane, 1 foot high, is found by experiment to lose ·5945 unit per square foot per hour, when heated 1° above the air in contact with it; but a high plane or wall loses less per square foot, for the following reason:—let Fig. 81 represent a plane 4 feet high, heated to 60°, while the air in contact with it is at 59°. The air in immediate contact with the wall being heated by it, is expanded thereby, becomes lighter than the surrounding air, and ascends in a constant current as shown by the arrows. Now for the first or lowest foot, the air is at 59° to begin with, but departs from it at a slightly increased temperature, so that there is *less* than 1° difference when it comes in contact with the second foot, and for that reason it receives less heat from it than it did from the lowest foot, and so throughout, each successive foot receives the air at progressively increased temperature and imparts to it less and less heat.

The loss by a vertical plane is given by the rule $A = \cdot361 + \frac{\cdot233}{\sqrt{H}}$, in which A = the loss in units per square foot per hour for 1° difference in temperature, and H = the height of

TABLE 74.—Of the VALUE of A for VERTICAL PLANES.

Height in Feet.	Units per Square Foot for 1° Difference.	Height in Feet.	Units per Square Foot for 1° Difference.
1	·5945	20	·4133
2	·5280	30	·4037
3	·4962	40	·3980
4	·4780	60	·3924
5	·4655	80	·3871
10	·4350	100	·3843

the plane or wall in feet. Table 74 has been calculated by this rule, and it shows that a wall 4 feet high, as in Fig. 81, loses ·4780 unit per square foot per hour, and by subtraction and analysis we may find the loss by each successive foot; thus a wall 2 feet high loses per Table 74, ·5280 per foot, or ·528 × 2 = 1·056 unit, and as the 1st foot loses ·5945 unit, the second must lose 1·056 – ·5945 = ·4615 unit; similarly, the third foot will lose (·4962 × 3) – (·5945 + ·4615) = ·4326 unit, and thus we obtain the numbers given in Table 75.

TABLE 75.—Of the LOSS of HEAT from CONTACT of COLD AIR by a WALL 4 feet high.

	Units per Square Foot per Hour for 1° Difference.
1st, or lowest foot	·5945
2nd foot	·4615
3rd ,, ...	·4326
4th ,, ...	·4234
Total	1·9120
Per square foot, mean	·4780

This explanation of the reason why a high wall does not lose the same amount of heat per square foot as a low one, will give some idea of the cause of the difference between bodies having the same area, but differing in form; those bodies lose most heat, whose form allows the most free access and circulation of the air which carries off their heat.

(273.) For a horizontal cylinder, the rule becomes A =

$\cdot 421 + \frac{\cdot 307}{r}$, in which A = the loss in units per square foot of surface per hour for a difference in temperature between the body and the air of 1°, and r = the radius of the cylinder in inches. Of course the cylinder is supposed to be of infinite length, so as to exclude the loss by the *ends*. The 2nd column in Table 76 is calculated by this rule. For a sphere, the rule becomes $A = \cdot 3634 + \frac{1 \cdot 0476}{r}$, in which A and r have the same signification as above, and thus we have col. 3 in Table 76.

TABLE 76.—Of the Loss of HEAT from CONTACT of AIR with HORIZONTAL CYLINDERS and SPHERES, per square foot per hour, for a difference in Temperature of 1° Fahr.

Diameter in Inches.	Value of A.	
	Horizontal Cylinder.	Sphere.
	Units.	Units.
2	·7280	1·4110
3	·6256	1·0618
4	·5745	·8872
5	·5440	·7824
6	·5230	·7126
7	·5087	·6627
8	·4978	·6253
9	·4892	·5962
10	·4824	·5729
12	·4722	·5380
18	·4551	·4798
24	·4466	·4507
36	·4381	·4219

(274.) "*Loss of Heat by Conduction.*"—Let P, Fig. 76, be a plate of building stone, 1 foot square, 1 inch thick, having its surface S maintained at 60°, and let the walls W W W and the air in contact with S be at 60° also, while the surface S′ is at 59°. There will then be no loss of heat by radiation or contact of air, because W and the air is at the same temperature as S, but a certain amount of heat will be transmitted through the material from S to S′; and for stone 1 inch thick, the loss will be 13·7 units per square foot per hour for 1° difference, as in our case. This amount will vary very greatly with the nature of the material, we will call it C, and its value is given by Table 77. The amount of heat also varies directly as the difference of temperature of the two surfaces S and S′, and inversely as the thickness, and hence we have the rule $C' = \frac{C \times d}{E}$, in which C' = the loss by conduction in units per square foot per hour; C = the conducting power of the material, E = the thickness of the plate in inches, and d = the difference of temperatures of S and S'. Thus a wall of stone 20 inches thick, having one surface at 70° and the other at 40°, taking the value of C at 13·7, will transmit $\frac{13\cdot7\times(70-40)}{20}=20\cdot55$ units per square foot per hour.

TABLE 77.—Of the VALUE of C, or the CONDUCTING POWER of MATERIALS, being the quantity of Heat in Units transmitted per square foot per hour, by a Plate 1 inch thick, the two surfaces differing in Temperature 1°. From the experiments of PÉCLET.

	Value of C.
Copper	515
Iron	233
Zinc	225
Lead	113
Marble, grey, fine-grained	28
,, white, coarse-grained	22·4
Stone, calcareous, fine	16·7
,, ,, ordinary	13·68
Glass	6·6
Baked Clay, brickwork	4·83
Plaster, ordinary	3·86
Oak, transmission perpendicular to the fibres	1·70
Walnut ,, ,, ,,	·83
Fir ,, ,, ,,	·748
Fir ,, parallel to the fibres	1·37
Walnut ,, ,, ,,	1·40
Gutta-percha	1·38
India-rubber	1·37
Brick-dust, sifted	1·33
Coke, pulverized	1·29
Cork	1·15
Chalk, in powder	·869
Charcoal of Wood, in powder	·636
Straw, chopped	·563
Coal, small, sifted	·547
Wood Ashes	·531
Mahogany Dust	·523
Canvas of Hemp, new	·418
Calico, new	·402
White Writing-paper	·346
Cotton Wool and Sheep Wool (any density)	·323
Eiderdown	·314
Blotting-paper, grey	·274

(275.) Fig. 77 represents a case in which the surface S loses heat *simultaneously* by Radiation, contact of Air, and Conduction. Here W, W, W, the air, and the surface S′ are all 1° lower than S, and the *total* loss is the sum of the separate losses in (266), (270), and (274), namely $U = \overset{R}{\cdot 736} + \overset{A}{\cdot 5495} + \overset{C}{13 \cdot 7} = 14 \cdot 9855$ units per square foot per hour.

(276.) If W, S′, and the air were not all of one and the same temperature, the case would be modified. Let H, in Fig. 78, be a cube of new cast-iron 1 foot square, heated to 100°, and placed in a room whose walls, B, B, are maintained at 40°, and let the internal air be at 60°; considering only the loss by one vertical side of the cube, the loss by radiation with new cast-iron will be by Table 71, = ·648 for 1° difference, therefore in our case $\cdot 648 \times (100 - 40) = 38 \cdot 88$ units, and the loss by contact of air will be $\cdot 5945 \times (100 - 60) = 23 \cdot 78$ units, so that the total loss is $38 \cdot 88 + 23 \cdot 78 = 62 \cdot 66$ units per square foot per hour.

(277.) But sometimes a body may be losing heat from one source, and at the same time receiving it from another; thus in Fig. 79, we have a cube of stone, say 3 feet square, whose temperature is maintained at 50°, placed in a room whose walls are at 40°, while the air is at 60°. Here the loss of heat by radiation is $\cdot 736 \times (50 - 40) = 7 \cdot 36$ units per square foot per hour; but by contact of air, heat will be *received*, not lost, the air being 10° warmer than the cube. By Table 74, a plane 3 feet high will lose or gain ·4962 unit per square foot per hour for 1°, therefore in our case the heat received will be $\cdot 4962 \times (60 - 50) = 4 \cdot 962$ units per square foot per hour, so that the final result is a loss of $7 \cdot 36 - 4 \cdot 962 = 2 \cdot 398$ units per square foot per hour. Here then we have a case in which a body is *simultaneously losing and receiving heat* from different sources.

(278.) These rules and tables are easily applied to practice; thus a wall of stone, 20 feet high and 50 feet long at 70°, exposed to air and all surrounding objects at 60°, will lose $(\overset{R}{\cdot 736} + \overset{A}{\cdot 4133}) \times 20 \times 50 \times 10 = 11493$ units per hour; a horizontal cylinder, of rusted cast-iron, 6 inches diameter and 40 feet long, having an area of $3 \cdot 14 \times \cdot 5 \times 40 = 62 \cdot 8$ square feet, will lose, for say 15° difference $(\overset{R}{\cdot 6868} + \overset{A}{\cdot 523}) \times 62 \cdot 8 \times 15 = 1140$ units per hour; and a sphere 2 feet diameter, having by the rules of mensuration an area of $3 \cdot 14 \times 4 =$

12·56.square feet, say painted with oil paint, will lose for 15° difference $(\overset{R}{\cdot 7583} + \overset{A}{\cdot 4507}) \times 12\cdot 56 \times 15 = 228$ units per hour.

(279.) "*The Loss of Heat by Buildings.*"—Having thus determined the value of the three elements R, A, and C, we will proceed to apply them to buildings artificially heated for the purposes of life, which is one of the most important subjects to which they are applicable. The heat lost by buildings may be divided into four portions; namely, that lost by the floor, by the ceiling, by the walls, and by the windows.

"*Loss by the Floor or Ground.*"—It is stated in (22) that the surface of the earth in ordinary cases varies in temperature with that of the air, but where the ground is covered by a building the case is different; it will then take pretty nearly the mean temperature of the earth at that place. We have shown (22) that at a depth of 20 feet the earth has a uniform and invariable temperature, having the mean yearly temperature of the air at that place; this in the case of London, is shown by Table 26 to be about 51°, and as this is pretty nearly the average temperature of our dwellings in the cold season, the ground or floor, being in contact with earth beneath and air above, both of that temperature, will take the same, and the loss of heat from this source will be nothing.

"*Loss by the Ceiling, &c.*"—When the room is covered with an ordinary lath-and-plaster ceiling, and that again protected from cooling influences by the roof, the loss will be so small that it may be neglected, and we shall consider in what follows that the loss of heat from this cause is nothing. But where there is no ceiling, and the roof itself is exposed to the heated air in the building, there will be a great loss of heat, but one exceedingly difficult to estimate.

(280.) "*Loss of Heat by the Walls.*"—We will take for illustration, the case of a room with brick walls and no windows, &c., as in Fig. 80, exposed on all sides to cooling influences, the external air and all the surrounding radiant objects having a temperature of 30°, while the internal air is maintained at 60°; and the problem to be solved is the amount of heat which will be received from the internal air, transmitted through the walls, and finally dissipated on the external objects.

To avoid complicating the question, we may assume that the room, Fig. 80, is of very large dimensions, so that the external surface has sensibly the same area as the internal surface. In practice this is near enough the truth in most cases, but it may perhaps be more correct to take the area at a mean between the two.

We are met at the outset with the difficulty that the temperatures of the *surfaces* of the walls are not known, in our case (and in most cases) we only know the temperature of the internal and external *air*, T and T'; the interior surface of the wall t must obviously have a temperature lower than the internal air T, otherwise it could not receive heat from it: similarly the exterior surface t' must have a higher temperature than the external air, &c., or it could not give out heat to it.

(281.) We will first calculate the temperature of the two surfaces of the wall t and t', and shall adopt the following general notation throughout:—

T = the temperature of the internal air.
T' = „ external air.
t = „ internal surface of wall.
t' = „ external surface of wall.
t'' = „ glass in windows, &c.
R = radiant power of the material, as per Table 71.
A = loss by contact of air, as per Tables 74, 76, &c.
C = conducting power of the material, as per Table 77.
$Q = R + A$.
U = units of heat per hour.
E = thickness of the wall in inches.

(282.) Taking the walls in our case, Fig. 80, at 40 feet high, we have by Table 74 the value of $A = {\cdot}398$; the value of R for brickwork we shall take at ·736, and C at 4·83, therefore $Q = 1{\cdot}134$, and we may determine the temperature of the internal surface of the wall (t) by the formula:—

$$t = \frac{Q(E\,.\,A\,.\,T + C\,.\,T') + A\,.\,C\,.\,T}{C\,(2\,A + R) + (E\,.\,A\,.\,Q)},$$ which in our case becomes

$$\frac{[1{\cdot}134 \times (14 \times {\cdot}398 \times 60 + 4{\cdot}83 \times 30)] + ({\cdot}398 \times 4{\cdot}83 \times 60)}{[4{\cdot}83 \times (2 \times {\cdot}398) + ({\cdot}736)] + (14 \times {\cdot}398 \times 1{\cdot}134)}$$

$= 48^{\circ}{\cdot}02$ the internal temperature of the wall.

(283.) Having found t, we may find t' by the formula:—

$$t' = \frac{C\,t + Q\,E\,T'}{C + Q\,E},$$ which in our case becomes

$$\frac{(4{\cdot}83 \times 48{\cdot}02) + (1{\cdot}134 \times 14 \times 30)}{4{\cdot}83 + (1{\cdot}134 \times 14)} = 34^{\circ}{\cdot}2,$$

which is the temperature of the external surface of the wall. Having thus found t and t', we may now calculate the quantity of heat transmitted by several formulæ.

(284.) Knowing the temperature of the internal air T, and of the internal surface of the wall in contact with it t, we can easily calculate the amount of heat which the wall will absorb, namely, $U = A \times (T - t)$, which in our case is $·398 \times (60 - 48·02) = 4·77$ units per square foot per hour.

(285.) We may also calculate from the known temperature of the two surfaces of the wall t and t', and the conducting power of the material C, irrespective of the temperatures of the internal and external air by the formula: $U = \frac{C \times (t - t')}{E}$, which in our case is $\frac{4·83 \times (48·02 - 34·2)}{14} = 4·77$ units per square foot per hour, as before.

(286.) We may also calculate from the known temperature of the external surface of the wall t', and the temperature of the external air and surrounding objects which absorb radiant heat, both being at 30°, by the formula: $U = Q \times (t' - T')$, in our case $1·134 \times (34°·2 - 30°) = 4·77$ units, as before.

(287.) These three formulæ show the steps of the cooling process; (284) shows that 4·77 units are absorbed from the internal air by the internal surface of the wall; (285) shows that the same amount of heat is transmitted in the same time from one surface of the wall to the other; and (286) proves that the heat thus received and transmitted to the outer surface, is finally dissipated by it on the external air, &c. It should be observed that the internal surface receives its heat only by contact of warm air, and none by radiation, because all the internal walls have one and the same temperature with the wall B, whose condition we have been examining; but the exterior surface loses its heat both by radiation and contact of air.

(288.) We may also determine the loss, if we know only t and T′, or the temperature of the *interior* surface of the wall, and of the *external air* by the rule $U = \frac{Q \times (t - T')}{1 + Q\frac{E}{C}}$, which in our case becomes $\frac{1·134 \times (48·02 - 30)}{1 + (1·134 \times \frac{14}{4·83}} = 4·77$ units as before. This is a useful rule for many cases, such as the wall of a furnace which has the temperature of the fire itself on one side, and is exposed to the air and radiant objects at low temperatures on the other side.

(289.) The loss of heat may also be calculated from the known temperatures of the external and internal *air* only; this is perhaps the most useful formula, because ordinarily these are the only known temperatures: $U = \frac{A.C.Q \times (T - T')}{C(2.A + R) + E.A.Q}$, which for our case becomes

$$\frac{\cdot 398 \times 4\cdot 83 \times 1.134 \times (60 - 30)}{[4\cdot 83 \times (2 \times \cdot 398) + (\cdot 736)] + (14 \times \cdot 398 \times 1\cdot 134)} = 4\cdot 77 \text{ units}$$

as before.

TABLE 78.—Of the VALUE of U, or the loss in Units per square foot per hour, by a Room exposed on all sides to Air, &c., 1° cooler than the internal Air (walls 40 feet high).

Brickwork.		Stone.	
Thickness.	U.	Thickness.	U.
Brick. Inches.		Inches.	
½ = 4½	·2312	6	·261
1 = 9	·1910	12	·234
1½ = 14	·1590	18	·212
2 = 18	·1404	24	·194
3 = 27	·1113	30	·179
4 = 36	·0920	36	·166

Table 78 has been calculated by these rules for stone and brickwork, for the latter we have taken C at 4·83, and for stone at 13·7, while R and A have the same value for both, namely, R = ·736, and A = ·398. Thus, take the case of a building, say 40 feet high, 100 feet long, and 60 feet wide, exposing 12,800 square feet, with internal air at 60° and external air at 30°; each square foot of brickwork, say 14 inches thick, will lose by the Table ·159 unit for 1°, or for 30° ·159 × 30 = 4·77 units per hour, and the total loss will therefore be 4·77 × 12800 = 61056 units. With stone walls 2 feet thick, the loss would have been ·194 × 30 × 12800 = 74496 units. For walls a little more or less than 40 feet high, the loss will be very nearly the same as per Table 78, and for ordinary cases the tabular numbers are correct enough for practice.

(290.) We have so far considered only the case of a building exposed on *all sides* to cooling influences. When only one face is exposed, as in the case of a room forming part of a large building, the case is somewhat different, and may be illustrated by Fig. 84, in which W is a room, only one wall of which is exposed to the external air and radiant objects at 30°, the internal air and surfaces of walls G, H, J, being maintained at 60°, and let F, F, F, &c., be other rooms. The interior surface of the wall B, whose condition we are now to consider, is therefore exposed not only to *air* at 60°, but also to radiation from the other walls at 60° also; it will therefore have a higher temperature than before, and will in consequence transmit more heat. Being exposed to similar influences on both sides, that is to say, being heated on one side by air and radiant bodies having one and the same temperature, and cooled on the other in the same manner, and the value of R and A being the same for both internal and external surfaces, the temperature of the wall *at the centre of its thickness* will be a mean between T and T′, or $\frac{60+30}{2} = 45°$, as in Fig. 83, which represents a portion of the wall B on a larger scale.

It will be seen by inspecting the figures, that this is analogous to the case of a wall 7 inches thick, with one surface maintained at 45°, while the other is exposed to air and radiant objects at 30°, as at O P, and we can calculate the amount of heat transmitted by our formula (288), which in our case becomes $\frac{1{\cdot}134 \times (45-30)}{1 + 1{\cdot}134 \times \frac{7}{4{\cdot}83}} = 6{\cdot}4$ units per square foot per hour. This is the amount transmitted from M to N, Fig. 83, and the same amount would pass from L to M because the conditions are the same, namely, the same thickness of wall, or 7 inches, and difference of temperature, or 15°; so that a wall 14 inches thick exposed to air and radiant objects at 60° on one side, and air and radiant objects on the other side at 30°, will transmit 6·4 units per square foot per hour, and this we have found without knowing the temperatures of the surfaces of the wall.

(291.) We can now find the temperatures of the two surfaces of the wall B; the internal surface has to receive 6·4 units, and as for 1° it would receive only 1·134 unit (namely, the value of Q), its temperature must be $\frac{6{\cdot}4}{1{\cdot}134} = 5°{\cdot}64$ lower than the internal air, &c., or $60° - 5°{\cdot}64 = 54°{\cdot}36$. Again, the external face has to lose 6·4 units, and must be 5°·64 warmer than the external air in order to do so, its temperature will therefore be $30° + 5°{\cdot}64 = 35°{\cdot}64$.

Knowing now the temperatures of the two faces of the wall B, its thickness, and conducting power C, we may verify these calculations by the formula (285), which in our case becomes $\frac{4\cdot83 \times (54^\circ\cdot36 - 35^\circ\cdot64)}{14} = 6\cdot4$ units as before. This is considerably more than the loss by a building exposed on all sides, which we found to be 4·77 units for the same internal and external temperature. Table 79 gives the loss in units per square foot per hour for buildings of brick and stone.

TABLE 79.—Of the Loss of HEAT per SQUARE FOOT per HOUR by BRICK and STONE WALLS, 40 feet high, in Buildings where only one face is exposed, and for 1° difference of Internal and External Temperature.

Brickwork.			Stone..	
Thickness.		U.	Thickness.	U.
Brick.	Inches.		Inches.	
½	= 4½	·371	6	·453
1	= 9	·275	12	·379
1½	= 14	·213	18	·324
2	= 18	·182	24	·284
3	= 27	·136	30	·257
4	= 36	·108	36	·228

(292.) It may appear anomalous that a building exposed on all sides should lose less heat per square foot than a wall in a room exposed to cooling influences on one side only. We assumed for our illustration in Fig. 84 that the temperature of the internal surfaces of the walls G, H, J, in the room W was 60°; but we have seen that the wall B borrows heat from them to the extent of $\cdot736 \times (60^\circ - 54^\circ\cdot36) = 4\cdot15$ units per square foot per hour; H will furnish the largest share, and we may assume it at one-half, and as the areas of H and B are equal, each square foot of H will give out to B $\frac{4\cdot15}{2} = 2\cdot075$ units per square foot per hour. But to do this, its surface K must be of higher temperature than H, and the difference of temperature by the rule $\frac{U \times E}{C} = (t - t')$ will be in our case $\frac{2\cdot075 \times 9}{4\cdot83} = 3^\circ\cdot86$; the temperature of K must therefore be $60^\circ + 3^\circ\cdot86 = 63^\circ\cdot86$; and as we may suppose that the heat received by K is derived from the air in the room F′, that air must be still warmer than K to the extent of $\frac{2\cdot075}{\cdot398} = 5^\circ\cdot2$; the temperature of the air in

F' will therefore be 63°·86 + 5°·2 = 69°·06. Thus it appears that to secure the conditions we assumed for the room W, with air and walls at 60°, a higher temperature is necessary in the rest of the building, and for that reason more heat is lost by the wall B than in a case like Fig. 80.

(293.) "*Loss of Heat by Glass in Windows, &c.*"—We will first take the case of a window in the room, Fig. 85, in which the interior walls and internal air in contact with the glass have one and the same temperature of 60°, and all the external radiant objects and external air have one and the same temperature of 30°. In that case, the glass being heated on one side and cooled on the other by similar influences, will have a temperature in the centre of its thickness (290), a mean between the two, or in our case $t'' = \frac{60 + 30}{2} = 45°$, and with thin glass we may assume that it has this temperature throughout. We may calculate the amount of heat received from within and dissipated without by the rule $U = Q \times (T - t'')$. For glass the value of R is by Table 71, ·5948, and the value of A by Table 74 for a window say 5 feet high, is ·4655; therefore $Q = \cdot5948 + \cdot4655 = 1\cdot0603$, and in our case the loss is $1\cdot0603 \times (60° - 45°) = 15\cdot9$ units per square foot per hour for 30° difference of internal and external temperature, or $\frac{15\cdot9}{30} = \cdot53$ unit for 1° by a window 5 feet high; for 10 feet and 20 feet the losses are ·515 and ·504 respectively.

(294.) We will now consider the loss by a window in a room like Fig. 80, in which the glass is exposed internally to air at 60° and radiant walls at 48°·02, while externally the temperature of both the air and radiant objects is 30°. In this case the temperature of the glass will not be a mean between T and T', but will be given by the formula $t'' = \frac{\frac{(T - t) \times A}{A + R} + t + T'}{2}$, which in our case becomes

$$\frac{\frac{(60° - 48°\cdot02) \times \cdot4655}{\cdot4655 + \cdot5948} + 48\cdot02 + 30}{2} = 41°\cdot65$$

This case is shown by Fig. 82; the heat received from within by contact of the air will be, $A \times (T - t'')$, or in our case $\cdot4655 \times (60° - 41°\cdot65) = 8\cdot41$; and by radiation, $= R \times (t - t'')$; in our case, $\cdot5948 \times (48°\cdot02 - 41\cdot65) = 3\cdot789$ units, making a total of $8 \cdot 541 + 3 \cdot 789 = 12 \cdot 3$ units per square foot per hour. It is obvious that the same amount of heat has to be dissipated on the external objects; here, however, the loss arises from radiation and contact of air both at the same temperature, and we have $U = (A + R) \times (t'' - T')$; in our case $(\cdot 4655 + \cdot 5948) \times (41^\circ \cdot 65 - 30^\circ) = 12 \cdot 3$ units, or the same as the heat received, proving that the temperature we found for the glass is correct. For 1° difference of internal and external air, the loss is $\frac{12 \cdot 3}{30} = \cdot 41$ unit per square foot per hour for a window 5 feet high; for 10 feet high and 20 feet the losses are ·4 and ·391 respectively.

(295.) "*Loss by Glass-houses, Conservatories, &c.*"—We will lastly consider the case of a glass-house exposed on all sides to air and radiant objects at 30°, the internal air being maintained at 60°, as in Fig. 86; here all the internal surfaces having the same temperature, all the heat received from the interior must be given out by the contact of heated air only; but the heat thus received is dissipated on external objects both by radiation and contact of air. The temperature of the glass will therefore not be a mean between T and T′, but will be found by the formula $t'' = \frac{(A \times T) + (A + R) \times T'}{(2 \times A) + R}$. Taking the height at 10 feet, our Table 74 gives ·435 for the value of A, and R being ·5948 as before, the rule becomes in our case—

$$\frac{(\cdot 435 \times 60^\circ) + (\cdot 435 + \cdot 5948) \times 30^\circ}{(2 \times \cdot 435) + \cdot 5948} = 38^\circ \cdot 9,$$

which is the temperature of the glass; the heat received from within will be $A \times (T - t'')$; in our case, $\cdot 435 \times (60^\circ - 38^\circ \cdot 9) = 9 \cdot 17$ units per square foot per hour; and the heat dissipated without will be $(A + R) \times (t'' - T')$, in our case $(\cdot 435 + \cdot 5948) \times (38^\circ \cdot 9 - 30^\circ) = 9 \cdot 17$ units also. For 1° difference of internal and external temperature we have therefore $\frac{9 \cdot 17}{30} = \cdot 306$ unit per square foot per hour.

(296.) Thus for the three cases we have investigated, we have found three different values of U under otherwise similar circumstances, namely, ·53 for a window in a room with only one face exposed; ·41 for a window in a brick building with 14-inch walls, exposed on all sides; and ·306 for a glass-house. Comparing the loss by glass with that by the 14-inch walls, we find in the first case $\frac{\cdot 53}{\cdot 213} = 2\cdot 5$, and in the second $\frac{\cdot 41}{\cdot 159} = 2\cdot 58$ times more heat lost by the glass than by the walls.

(297.) The foregoing facts and results will enable us to explain some anomalies which are matters of common observation, but the philosophy of which is not generally understood. Thus, for instance, it is well known that a room warmed by an open fire is much more comfortable to the occupants than if heated by steam or hot-water pipes. Fig. 80 may illustrate this case; say that the air is heated by steam-pipes to 60°, and imparts its heat to the walls and the window, both are therefore of necessity colder than the air, and our figure shows that the walls are about 12°, and the glass 19°, colder than the air. The occupant is therefore suffering from the abstraction of his animal heat by radiation of walls at 48° and windows at 41°, while the air is at the comfortable temperature of 60°, and if we would raise the *walls* to 60°, it could only be done by raising the temperature of the air to about 75°, which would be oppressive and injurious.

(298.) When a room is warmed by an open fire, the walls are warmer than the air, for radiant heat has the remarkable property of passing through air (in moderate thickness) without raising its temperature (268). By a common open fire, radiant heat only is given out usefully, for all the heated air passes up the chimney and is practically lost, and the radiant heat, passing through the air without sensible loss, is absorbed by the walls, and raises their temperature. The air enters the room by the crevices in the door-way and windows, at the external temperature, and is heated by contact with the walls, but must obviously be always colder than the walls by which it is heated, so that the occupant breathes air refreshingly cool, while the walls being comparatively highly heated, do not absorb his animal heat with inconvenient rapidity.

It will be evident from this, that to obtain a comfortable temperature the walls themselves must be heated rather than the air. This may be done by causing hot air to pass by channels formed in the walls, see (353) and Fig. 110. The same result may be obtained by open fires or stoves highly heated. But in the last case, the air heated by the outside of the stove should pass away up the chimney without mixing with the air in the room, so that the radiant heat alone would be utilized, and half the heating power of the coals would be wasted.

CHAPTER XIII.

LAWS OF COOLING AT HIGH TEMPERATURES.

(299.) With high temperatures and great differences of temperature, the simple formulæ we have given require correction, as we have stated in (267) and (271).

"*Loss by Radiation.*"—The rule in (267) assumes that the loss by radiation is simply proportional to the *difference* of temperature of the radiant body and the absorbant; but Dulong has shown that with the same difference the loss varies with the *absolute temperature* of the absorbent, so that, for instance, if in Fig. 74 the temperature of W W had been 212° and of S = 213°, the loss of heat per degree would have been about double the amount with the respective temperatures 60° and 59°. The loss of heat increases also in a much more rapid ratio than the difference of temperature, thus, with 432° difference, and with the absorbent at 212° (the Radiant being in that case at 644°), the loss per degree is six times greater than at low temperatures as in (267). Dulong has given rules which agree well with experiment, up to a difference in temperature of 468°. This rule is a very difficult one to apply, but it may be put in such a form as to give a *ratio* by which calculations by the simple rule may be easily corrected. The rule then becomes

$$\frac{124\cdot72 \times 1\cdot0077^{t} \times (1\cdot0077^{T} - 1)}{T} = R'',$$

in which t = the temperature of the absorbent, or recipient of radiant heat in degrees centigrade; T = the *excess* of temperature of the radiating body in degrees centigrade; and R″ = the *ratio* of loss of heat under the given temperatures. Table 80 has been calculated by this rule; but the temperatures are reduced to Fahrenheit's scale. The constant 124·72 is given by Péclet, who found the rule to agree perfectly with his own experiments. The loss of heat at *very* high temperatures, as calculated by this rule, becomes exceedingly great, as is shown by Table 82. In all probabilty there is considerable error in applying the rule to such extreme cases; but we obtain by it a nearer approximation to the truth than we could get without its assistance. This Table shows that with a radiant body at a clear red heat of 1860°, the loss is about 300 ! times the amount due by the simple formula, and at a bright white-heat of 2580°,

Table 80.—For Radiant Heat; being the *ratio* of Heat emitted or absorbed at Different Temperatures, according to the Experiments and Formula of Dulong.

Excess of Temperature of the Radiant above that of the Recipient in degrees Fahr.	Temperature of the Recipient of Radiant Heat, in Degrees Fahr.											
	32°	50°	59°	68°	86°	104°	122°	140°	158°	176°	194°	212°
	Ratio of Heat Emitted or Absorbed.											
18°	·997	1·075	1·120	1·165	1·254	1·355	1·467	1·580	1·702	1·848	1·994	2·150
36	1·032	1·083	1·160	1·206	1·299	1·403	1·520	1·680	1·763	1·914	2·064	2·227
54	1·071	1·155	1·200	1·251	1·348	1·449	1·576	1·70	1·83	1·985	2·142	2·310
72	1·115	1·202	1·250	1·302	1·403	1·515	1·640	1·760	1·900	2·066	2·230	2·400
90	1·163	1·254	1·310	1·360	1·463	1·580	1·710	1·840	1·980	2·154	2·330	2·510
108	1·212	1·307	1·360	1·416	1·525	1·648	1·784	1·920	2·070	2·280	2·423	2·615
126	1·26	1·36	1·42	1·48	1·59	1·72	1·86	2·00	2·16	2·34	2·52	2·72
144	1·32	1·42	1·48	1·54	1·65	1·79	1·94	2·08	2·24	2·44	2·64	2·83
162	1·37	1·48	1·54	1·60	1·73	1·86	2·02	2·17	2·34	2·54	2·74	2·96
180	1·44	1·55	1·61	1·68	1·81	1·95	2·11	2·27	2·46	2·66	2·87	3·10
198	1·50	1·62	1·69	1·75	1·89	2·04	2·21	2·38	2·56	2·78	3·00	3·24
216	1·58	1·69	1·76	1·83	1·97	2·13	2·32	2·48	2·68	2·91	3·13	3·38
234	1·64	1·77	1·84	1·90	2·06	2·28	2·43	2·52	2·80	3·03	3·28	3·46
252	1·71	1·85	1·92	2·00	2·15	2·33	2·52	2·71	2·92	3·18	3·43	3·70
270	1·79	1·93	2·01	2·09	2·22	2·44	2·64	2·84	3·06	3·32	3·58	3·87
288	1·89	2·03	2·12	2·20	2·37	2·56	2·78	2·99	3·22	3·50	3·77	4·07
306	1·98	2·13	2·22	2·31	2·49	2·69	2·90	3·12	3·37	3·66	3·95	4·26
324	2·07	2·23	2·33	2·42	2·62	2·81	3·04	3·28	3·53	3·84	4·14	4·46
342	2·17	2·34	2·44	2·54	2.73	2·95	3·19	3·44	3·70	4·02	4·34	4·68
360	2·27	2·45	2·56	2·66	2·86	3·09	3·35	3·60	3·88	4·22	4·55	4·91
378	2·39	2·57	2·68	2·79	3·00	3·24	3·51	3·78	4·08	4·42	4·77	5·15
396	2·50	2·70	2·81	2·93	3·15	3·40	3·68	3·97	4·28	4·64	5·01	5·40
414	2·63	2·84	2·95	3·07	3·31	3·51	3·87	4·12	4·48	4·87	5·26	5·67
432	2·76	2·98	3·10	3·23	3·47	3·76	4·10	4·32	4·61	5·12	5·53	6·04

it rises to 4604 !! times that amount. This may be very incorrect; but the extreme rapidity with which a body at white-heat cools down to orange and cherry red, &c., seems to indicate that at extreme temperatures the loss of heat is exceedingly rapid.

(300.) The application of Table 80 is very simple. Say we have a mass of wrought-iron heated to 600° in a chamber whose walls are at 190°. By Table 71, the radiant power of an ordinary surface of wrought-iron is ·5662, and by the simple rule we have $\cdot 5662 \times (600 - 190) = 232$ units per square foot per hour; but by Table 80, the nearest number to the temperature of the absorbent is 194°, and to the difference or ($600 - 190 = 410°$), is 414°, and the ratio for those two numbers is by the Table = 5·26, and the true loss by radiation in our case is therefore $232 \times 5\cdot 26 = 1220$ units per square foot per hour.

(301.) "*Loss by Contact of Air.*"—The researches of Dulong show that the loss by contact of cold air is independent of the *absolute* temperature of the heated body, differing in this respect from radiant heat; but he found that the heat lost increases more rapidly than the simple ratio of excess of temperature. Putting his formula, with the constants given by the experiments of Péclet, in such a form as to give us a *ratio* for the different temperatures, we have the rule $\frac{\cdot 552 \times t^{1\cdot 233}}{t} = R'''$, in which t = the difference of temperature of the body and the air in contact with it in degrees centigrade, and R''' = the ratio of loss of heat with that difference; and thus we obtain the numbers in Table 81 and in col. 5 of Table 82. It will be observed that the departures from the simple law are much less than with radiant heat; at the extreme temperature of 2580°, and with air at 59°, the loss is only 2·985 times the amount given by the simple rule.

Applying this to the case in (300), say that the heated body was a vertical plane 4 feet high, Table 74 gives ·478 for the value of A, and with the air at 190°, we have by the simple rule $\cdot 478 \times (600 - 190) = 196$ units; but for a difference of 410°, the nearest number in Table 81 is 414°, for which the ratio is 1·96, and hence we have $196 \times 1\cdot 96 = 384$ units per square foot per hour. Adding the respective losses by radiation and contact of air together, we obtain $1220 + 384 = 1604$ units as the total loss per square foot per hour.

(302.) We may apply these rules to the case of a steam-pipe. Say the air of the room and the walls are at 60°, and the steam-

TABLE 81.—Of the *Ratio* of HEAT EMITTED or ABSORBED by CONTACT of AIR with given DIFFERENCES of TEMPERATURE.

Difference of Temperature of the Air and the Body in Contact.	Ratio of Heat.	Difference of Temperature of the Air and the Body in Contact.	Ratio of Heat.
18°	·94	252°	1·742
36	1·11	270	1·774
54	1·22	288	1·800
72	1·305	306	1·827
90	1·372	324	1·852
108	1·433	342	1·874
126	1·486	360	1·897
144	1·533	378	1·920
162	1·575	396	1·940
180	1·615	414	1·960
198	1·650	432	1·980
216	1·684	450	2·000
234	1·720		

pipe at 210°, or 150° difference. Then the correction for radiant heat by Table 80 is say 1·5, and for contact of air by Table 81 = 1·55. Table 76 gives for horizontal cylinders of 2, 3, 4, and 6 inches diameter, the respective values of A at ·728, ·6256, ·5745, and ·523, and we have

	R. Diff. Ratio.	By Radiation.	A. Diff. Ratio.	By Contact.	Units per Sq. Ft. per Hour.
For 2 in. diam.	(·7 × 150 × 1·5) =	157·5, and	(·728 × 150 × 1·55) =	169·5	= 327
3 ,,	,,	,,	(·6256 × 150 × 1·55) =	145·5	= 303
4 ,,	,,	,,	(·5745 × 150 × 1·55) =	133·5	= 291
6 ,,	,,	,,	(·523 × 150 × 1·55) =	121·5	= 279

The weight of steam condensed to water at 212° per hour will be found by dividing the units of heat by 966 (the latent heat of steam), and we thus obtain ·338, ·312, ·301, and ·289 lb. of water per square foot per hour.

(303.) "*Enclosed Pipes, &c.*"—It should be observed that these calculations apply strictly to the case of a pipe freely exposed to air and radiant walls, &c., both of the same temperature, and this is nearly true where pipes are fixed in the room to be heated; but where they are enclosed in small channels under the floor, the case is very different, for in that case the walls enclosing the pipe become highly heated, and were they not continuously cooled by the air passing through them, they would soon take the same temperature as the pipe itself,

TABLE 82.—Of the RATIO of LOSS of HEAT at VERY HIGH TEMPERATURES, by the Formulæ of DULONG.

Temperature of the Heated Body.	Temperature of the Air in Contact with the Body, and of all Surrounding Objects.	Difference of Temperature of the Body, and of the Air, and Surrounding Objects.	Ratio of Heat lost at different Temperatures by	
			Radiation.	Contact of Air.
490°	60°	430°	3·10	1·980
600	,,	540	4·19	2·085
780	,,	720	7·17	2·230
960 Red, just visible ..	,,	900	12·68	2·348
1140	,,	1080	23·01	2·450
1320 Dull red	,,	1260	42·70	2·540
1500 Dull cherry red ..	,,	1440	80·67	2·620
1680 Cherry red	,,	1620	154·5	2·693
1860 Clear red	,,	1800	299·7	2·760
2220 Clear orange	,,	2160	1159·0	2·880
2580 White, bright ..	,,	2520	4604·0	2·985

NOTE.—For the loss at lower Temperatures, see Tables 80 and 81.

radiation from the pipe would cease altogether, and it would give out only about half the amount of heat. Taking the 4-inch pipe as an example from the cases just calculated, the loss by radiation is ·7 × 150 × 1·5 = 157·5 units, or rather more than half of the total heat emitted, which was 291 units. But the heat received by radiation from the pipe is given out again by the wall to the air, and the temperature of the wall rises only until the two are equal to one another. This temperature we find by trial to be about 158°. The wall will then be 210 − 158 = 52° colder than the pipe, and the ratio by Table 80 being 1·83 for a recipient at 158°, and the absorbent powers of brickwork being by Table 71 ·736, we have ·736 × 52 × 1·83 = 69·7 units received from the pipe. The amount given out by the wall to the air may be found by taking the value of A for a wall say 2 feet high from Table 74 at ·528 and the ratio from Table 81 for 158° − 60° = 98° difference at say 1·39, and we have ·528 × 98° × 1·39 = 71·9 units, or nearly the same amount as was received, showing that the temperature of 158° is nearly correct. The pipe at 210°, exposed to absorbing walls at 158°, and air at 60°, will lose (·7 × 52 × 1·83) + (·5745 × 150 × 1·55) = 200·2 units instead of 291 units, as we found for the case of air and walls of the same temperature; the ratio is $\frac{200\cdot2}{291} = \cdot688$, or say 70 per cent.; see also (382.)

(304.) "*Effect of Polished Metal Surfaces, &c.*"—The amount of heat lost by contact of air is not affected by the nature of the surface of the body, but that lost by radiation varies exceedingly, and the sum total is modified considerably. This may be illustrated by taking the case of a horizontal pipe 4 inches diameter outside, as in the last examples, heated at 210°, with air at 60°, &c., but with varying character of radiating surface. Taking the values of R from Table 71, and A as before from Table 76, at ·5745, we have the results shown by Table 83.

It will be observed, that the difference between a pipe in its ordinary state and a whitewashed one is very small indeed; blackening with lamp-black increases the loss about 11 per cent.; and a blackened pipe loses $\frac{317\cdot97}{139\cdot55} = 2\cdot27$ times as much heat as one of silvered copper highly polished like a mirror. Where it is desired to lose as little heat as possible, tinned iron (or common tin-plate) is a very effective and cheap material, losing less than half the amount of a blackened surface.

(305.) "*Effect of Thickness of Metal in Pipes heated internally.*"—We have assumed in the preceding calculations, and elsewhere throughout the work, that the outside of a steam-pipe has the same temperature as the steam, &c., inside, and this is practically true with thin pipes, such as are commonly used. With great thicknesses, however, the external temperature becomes sensibly less than that of the internal steam; it will be interesting to investigate the case generally.

We cannot calculate the loss of heat by the ordinary rules (274), because the surface dissipating the heat has a greater area than the surface receiving it; the ordinary rules suppose the wall, &c., transmitting heat to have parallel plain surfaces and to have a very large (or infinite) area, so that the heat may be considered as travelling in parallel lines from surface to surface, but in a thick pipe the heat travels in radial lines, and the amount transmitted cannot be calculated in the common way.

Let r = the radius of the inside of the pipe, in inches,
r' = " outside "
R = the radiant power of the outside surface, Table 71.

TABLE 83.—Of the Loss of HEAT by a HORIZONTAL PIPE 4 inches outside Diameter, heated to 210° and exposed to Air, &c., at 60°, showing the effect of different kinds of Radiating Surface.

KIND OF SURFACE.	R.	Diff.	Ratio.	By Radiation. Units.	A.	Diff.	Ratio.	By Contact of Air. Units.	Total per Hour per Square Foot. Units.	Ratio, Cast-iron rusted=1.
Silvered Copper, polished ..	(·02657 ×	150 ×	1·5) =	5·98 +	(·5745 ×	150 ×	1·55) =	133·57	139·55	·4860
Tinned Iron, do. ..	(·08585 ×	do.	) =	19·32 +		do.		do.	152·89	·5325
Iron, Sheet, do. ..	(·092 ×	do.	) =	20·7 +		do.		do.	154·27	·5373
Whitewashed	(·6786 ×	do.	) =	152·7 +		do.		do.	286·27	·9971
Cast-iron, rusted	(·6868 ×	do.	) =	154·53 +		do.		do.	287·10	1·0000
Lamp-blacked	(·8196 ×	do.	) =	184·4 +		do.		do.	317·97	1·1070

NOTE.—The loss by pipes of other sizes, and by bodies of other forms will not differ very materially from the numbers given for a 4-inch horizontal pipe; so that this Table may be used with approximate correctness for other cases. The difference between a whitened and a blackened pipe is shown to be less than is commonly supposed; the Table gives the loss of a whitewashed pipe at $\frac{286\cdot27}{317\cdot97} = \cdot9$, or 90 per cent. of the amount lost by a lamp-blacked one.

Let A = the loss of horizontal cylinder by contact of air (273), and Table 76.
Q = R + A.
C = conducting power of the material of the pipe, Table 77.
N = (log. r' − log. r) × 2·3.
t = temperature of the steam, &c., and of inside surface.
T′ = „ external air, and radiant objects.
U = units of heat lost per square foot per hour.
U′ = „ per foot run per hour.

$$\text{then } \frac{\cdot 5233 \times Q \times r' \times C \times (t - T')}{C + (Q \times r' \times N)} = U'.$$

Taking the case of a pipe 4 inches bore, 4 inches thick in cast-iron, 12 inches outside diameter, with steam at 212° and air, &c., at 60° or 152° difference, we may take R at ·7, A at ·4722, Q at 1·1722, and C at 233, for N we have log. of 6 = ·778, and log. of 2 = ·301, and N = (·778 − ·301) × 2·3 = 1·097, and the rule becomes $\frac{\cdot 5233 \times 1\cdot 1722 \times 6 \times 233 \times 152}{233 + (1\cdot 1722 \times 6 \times 1\cdot 097)}$ = 541·5 units per foot run; and a pipe 12 inches or 1 foot diameter, having an area of 3·14 square feet per foot in length, this is equal to $\frac{541\cdot 5}{3\cdot 14}$ = 172·4 units per square foot per hour.

(306.) But with a good conductor of heat, such as cast-iron, the external temperature of the pipe will be high, and these numbers will require correction (299) (301) by Tables 80 and 81, and to apply these we require to know the temperature of the external surface.

By the rule in (286) U = Q × (t − T′), and hence $\frac{U}{Q}$ = (t − T′), which in our case becomes $\frac{172\cdot 4}{1\cdot 1722}$ = 147° above the atmosphere; the temperature of the outer surface must therefore be 147 + 60 = 207°, or 5° less than that of the inside of the pipe. With that temperature, the correction by Table 80 is 1·48, and for A by Table 81 1·535, and calculating the true loss as in (302) we have (·7 × 1·48) + (·4722 × 1·535) × 147 = 258·8 units per square foot per hour, or 258·8 × 3·14 = 812·6 units per foot run.

With a pipe infinitely thin in metal: A becomes ·5745, Q 1·2745, C is cut out, and N being 0, the expression (Q × r' × N) vanishes, and the rule becomes ·5233 × Q × r' × (t − T′), or in our case ·5233 × 1·2745 × 152 = 202·77 units per foot run, or the area of a 4-inch pipe being $\frac{3{\cdot}14 \times 4}{12} = 1{\cdot}0467$ square foot, the loss is $\frac{202{\cdot}77}{1{\cdot}0467} = 193{\cdot}7$ units per square foot.

This might have been calculated by the common rule, $Q \times (t - T') = U$, or in our case 1·2475 × 152 = 193·7 units per square foot per hour, or the same as by the other rule.

Correcting these numbers by Tables 80 and 81, we have the true loss for 152° difference = (·7 × 1·5) + (·5745 × 1·55) × 152 = 294·9 units per square foot, or 294·9 × 1·0467 = 308·7 units per foot run per hour; so that the pipe 4 inches thick loses $\frac{812{\cdot}6}{308{\cdot}7} = 2{\cdot}6$ times as much heat per foot run, as one infinitely thin.

TABLE 84.—Of the Loss of HEAT by a HORIZONTAL CAST IRON PIPE, 4 inches bore, with different Thicknesses of Metal, heated with steam inside at 212°, and freely exposed to air, &c., at 60°.

Diameter of the Pipe in inches.		Thickness of Metal in inches.	Loss of Heat per Hour.			Temperature.		
			Per foot run.		Per Square Foot. Units.	Of the External Surface.	Of the Internal Surface.	Difference.
Inside.	Outside.		Units.	Ratio.				
						°	°	°
4	4	0	308·7	00	294·9	212	212	·000
4	6	1	441·17	1·4291	281·0	211·022	212	·978
4	8	2	571·4	1·851	273·0	210	212	2·0
4	12	4	812·6	2·600	258·8	207	212	5·0
4	16	6	1021·6	3·300	244·0	191	212	21·0

Table 84 has been calculated in this way; we find that even with the great thickness of 4 inches, the external temperature is only 5° less than that of the steam; with a thickness of 1 inch the difference is less than 1°, and of course with a pipe of the ordinary thickness of $\frac{5}{16}$ to $\frac{1}{2}$ inch, the external temperature will be sensibly the same as the internal, and the assumption that the heat lost is proportional to the external diameter is practically correct.

(307.) "*Steam Pipes, &c., cased in Bad Conductors of Heat.*"—The loss of heat by naked pipes to steam-engines, &c., is very considerable, and where the length is great it becomes serious, not only from the waste of fuel, but from the formation of water by condensation, which is obstructive to the working of the engine. Thus, with 35 lbs. steam, having by Table 11 a temperature of about 280°, the loss of heat by Table 66 with a 4-inch pipe would be 587 units per foot run per hour, and as by (48) a horse-power requires about 70,000 units, we find that a 4-inch steam-pipe, $4\frac{5}{8}$ diameter outside and 100 feet long, would lose $\frac{587 \times 100}{70{,}000} = {\cdot}84$-horse power. This loss may be greatly reduced by casing the pipes in a material which conducts heat badly: see Table 77. The woollen felt which is made for this special purpose is the best and cheapest material.

Adopting the same notation as before (305), but putting C for the conducting power of the casing, and r'' for its outer radius, N will be $(\log. r'' - \log. r') \times 2\cdot3$, and the rule becomes

$$\frac{\cdot5233 \times Q \times r'' \times C \times (t - T')}{C + (Q \times r'' \times N)} = U'.$$

For illustration of the effect of casing with different materials, we will take the case of a pipe 4 inches outside diameter, heated to 212°, the casing being covered in all cases with canvas, so as to give the same radiating power to the outer surface, and thus exhibit the variation in loss of heat due to conducting power alone.

	Thickness of Casing in Inches.					
	C.	½-inch.	1 inch.	2 inches.	4 inches.	6 inches.
Woollen Felt, or Cotton Wool	·323	79·5	52·4	34	22·5	18·1
Sawdust	·523	108	76·7	52	35·6	28·92
Fir-wood (transmission perpendicular)	·748	131	99	71	49·6	40·7
Coal Ashes (coke pulverized)	1·29	165	138	108	80·6	67·6
Plaster	3·86	216	216	209	189·0	172·0
Stone	13·68	244	272	316	367·0	388·0
Marble, grey, fine	28·00	250	287	352	452·0	519·0

Thus, for instance, with a casing of fir-wood, 1 inch thick, the outer diameter becomes 6 inches, $r'' = 3$ inches, R by Table 71 for canvas or calico $= {\cdot}7461$, A by Table 76 $= {\cdot}523$, therefore $Q = 1\cdot2691$, and $C = {\cdot}748$ by Table 77. For N we have $\log. 3 = {\cdot}477$ and $\log. 2 = {\cdot}301$, and $N = ({\cdot}477 - {\cdot}301) \times 2\cdot3 = {\cdot}405$. Then the rule becomes—

$\frac{\cdot 5233 \times 1\cdot 2691 \times 3 \times \cdot 748 \times 152}{\cdot 748 + (1\cdot 2691 \times 3 \times \cdot 405)} = 98\cdot 9$, or say 99 units per foot run per hour.

Calculating in this way with the different conducting powers and thicknesses, we obtain the numbers in the preceding Table.

(308.) With low conductors such as woollen felt, and considerable thicknesses such as 4 or 6 inches, these numbers are correct enough for practice, but with thin casings and better conductors, the external temperature will be high, and they require correction by Tables 80 and 81. We must first find the temperature: continuing the case with fir-wood, in which we have a loss of 99 units per foot run, the area of a 6-inch cylinder is $\frac{3\cdot 14 \times 6}{12} = 1\cdot 57$ square foot, and the loss of heat per square foot is $\frac{99}{1\cdot 57} = 63$ units, and the excess of temperature being $\frac{\mathrm{U}}{\mathrm{Q}} = (t - \mathrm{T}')$, becomes in our case $\frac{63}{1\cdot 2691} = 50^\circ$ above the external air, and the temperature of the external surface is $50 + 60 = 110^\circ$. With 50° excess, the correction of R by Table 80 is 1·2, and of A by Table 81 = 1·22, and the true loss of heat is $(\cdot 7461 \times 1\cdot 2) + (\cdot 523 \times 1\cdot 22) \times 50 = 76$ units per square foot, or $76 \times 1\cdot 57 = 119$ units per foot run per hour.

Comparing this with the loss by a 4-inch pipe uncased, but still covered with thin canvas, the corrections for 152° difference by Tables 80 and 81 are respectively 1·5 and 1·55 as in (302), and we have $(\cdot 7461 \times 1\cdot 5) + (\cdot 5745 \times 1\cdot 55) \times 152 = 305\cdot 4$ units per square foot, or $305\cdot 4 \times 1\cdot 0467 = 320$ units per foot run, the ratio is $\frac{119}{320} = \cdot 372$, the uncased pipe being 1·0.

Table 85 has been calculated in this way throughout: it will be observed that with low conductors, such as woollen felt, the loss becomes rapidly less with increase in thickness, but with such good conductors as stone and marble, the loss with all thicknesses is *greater* than by an uncased pipe, and the thicker the casing, the greater the loss; it is therefore worse than useless to use even moderately good conductors for such a purpose, and it is to illustrate this fact that such materials are introduced in the Table.

(309.) The effect of a given thickness of casing is not quite the same for all diameters of pipe; Table 86 gives the loss of heat by horizontal pipes of different diameters from 2 to 12

TABLE 85.—Of the Loss of HEAT by a HORIZONTAL PIPE 4 inches OUTSIDE cased in different Thicknesses of various Materials and covered with Thin the Loss

Material of the Casing.	Conducting Power of the Casing (C.)	THICKNESS OF CASING ON A							
		½-inch thick.				1 inch thick.			
		Loss of Heat per hour.			Temp. of the External Surface.	Loss of Heat per hour.			Temp. of the External Surface.
		Per Foot run.		Per Foot square, Units.		Per Foot run.		Per Foot square, Units.	
		Units.	Ratio.			Units.	Ratio.		
Woollen Felt, or Cotton Wool	·323	93	·2900	71	107°	56	·175	35·7	86°
Saw-dust (Mahogany) ..	·523	131	·409	100	123	86	·269	54·9	98
Fir-wood (transmission against grain)	·748	166	·519	127	137	119	·372	76·0	110
Coal Ashes (Coke pulverized)	1·29	225	·703	172	158	173	·541	110	129
Plaster, common	3·86	313	·978	239	188	299	·934	191	168
Stone	13·68	365	1·14	279	204	407	1·272	259	197
Marble, grey, fine	28·00	376	1·175	287	208	430	1·344	274	204

NOTE.—The ratio in this Table gives the loss by the cased pipe, compared with the wood casing, 2 inches thick, the loss per foot run is ·238, that by an uncased pipe

inches, cased with from ¼ inch to 6 inches of felt, &c. The low temperature of the external surface with considerable thicknesses is remarkable; thus a 4-inch pipe with 6 inches of casing has a surface temperature of 63°·58, or 3°·58 only above the atmosphere. The ratio in this Table gives the comparative losses with cased and *naked* pipes (not covered with canvas as in Table 85); this is obtained by the same method as in (302), and thus we have—

$$\begin{array}{lllll} & \text{R. Ratio.} & \text{A. Ratio.} & \text{Diff.} & \text{Circum. in feet.} \quad \text{Units.} \\ \text{For 2-inch pipe} & (\cdot 7\times 1\cdot 5)+ & (\cdot 728\ \times 1\cdot 55) & \times 152\times & \cdot 5236 = 173\cdot 4 \\ \quad 4 \quad ,, & \quad ,, & (\cdot 5745\times 1\cdot 55) & \times 152\times & 1\cdot 0467 = 308\cdot 7 \\ \quad 8 \quad ,, & \quad ,, & (\cdot 4978\times 1\cdot 55) & \times 152\times & 2\cdot 094\ = 576\cdot 5 \\ \quad 12 \quad ,, & \quad ,, & (\cdot 4722\times 1\cdot 55) & \times 152\times & 3\cdot 142\ = 851\cdot 0 \end{array} \left.\vphantom{\begin{array}{l}a\\a\\a\\a\end{array}}\right\} \text{Per foot run.}$$

Table 86 shows that a 4-inch pipe cased in 2 inches of felt loses 34·7 units per foot run, but we see that the naked pipe would have lost 308·7 units, or $\frac{308\cdot 7}{34\cdot 7} = 8\cdot 9$ times as much as

AMETER, heated to 212°, freely exposed to Air and Radiant Objects at 60°, nvas; showing the effect of the varying Conducting Power of the Casing on Heat.

E 4 INCHES DIAMETER OUTSIDE.

2 inches thick.				4 inches thick.				6 inches thick.			
oss of Heat per hour.			Temp. of the External Surface.	Loss of Heat per hour.			Temp. of the External Surface.	Loss of Heat per hour.			Temp. of the External Surface.
er Foot run.		Per Foot square, Units.		Per Foot run.		Per Foot square, Units.		Per Foot run.		Per Foot square, Units.	
nits.	Ratio.			Units.	Ratio.			Units.	Ratio.		
4·7	·1086	16·6	73°	22·36	·070	7·124	65°·88	18·10	·0566	4·323	63°·58
4·8	·1725	26·2	80	35·86	·112	11·42	69·27	28·92	·0904	6·908	65·73
6·1	·2380	36·4	87	51·15	·160	16·29	73	41·06	·1283	9·81	68·04
4	·3875	59·5	101	85·7	·268	27·3	81	69	·2156	16·6	73·4
2	·8500	130	140	223	·697	71·0	109	201	·628	48	95
0	1·407	215	181	499	1·56	159	156	498	1·55	119	137
9	1·622	248	195	644	2·01	205	178	720	2·25	172	163

s by a pipe without casing, but covered with canvas like the others; thus with fir ing 1·0, &c. The uncased pipe loses 320 units per foot run.

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the cased pipe. This ratio is given for each diameter and thickness in the Table, and may be used with sufficient correctness to find the loss by other sizes of pipes: thus with 2 inches of felt, the ratio for a 4-inch pipe is 8·9, and for 8-inch 9·71, a 6-inch one might therefore be taken at $\frac{8\cdot9 + 9\cdot71}{2} = 9\cdot3$. Then by Table 66 the loss by a 6-inch naked pipe at 210° is 493 units per foot run; cased in 2 inches of felt, this would be reduced to $\frac{493}{9\cdot3} = 53$ units, &c. It will be observed that very thin casings of felt are effective in reducing the loss of heat; thus $\frac{1}{4}$-inch reduces the loss to less than half the amount with a naked pipe, &c. Great thicknesses are very effective, but they become unsightly from the great increase in bulk; one inch is a common thickness, and the loss is then about $\frac{1}{5}$th of that with a naked pipe.

TABLE 86.—Of the Loss of HEAT by HORIZONTAL PIPES of different Diameters cased in WOOLLEN FELT or COTTON WOOL, and covered with Thin Canvas, &c.; the Pipes being heated to 212°, and freely exposed to External Air and Radiant Objects at 60°.

Thickness of the Felt or Cotton-wool Casing in inches.	DIAMETER OF THE PIPE (OUTSIDE, WITHOUT CASING).															
	2 inches Diameter.				4 inches Diameter.				8 inches Diameter.				12 inches Diameter.			
	Loss of Heat per hour.			Temp. of the External Surface.	Loss of Heat per hour.			Temp. of the External Surface.	Loss of Heat per hour.			Temp. of the External Surface.	Loss of Heat per hour.			Temp. of the External Surface.
	Per Foot run.		Per Foot square, Units.		Per Foot run.		Per Foot square, Units.		Per Foot run.		Per Foot square, Units.		Per Foot run.		Per Foot square, Units.	
	Units.	Ratio.			Units.	Ratio.			Units.	Ratio.			Units.	Ratio.		
0	173·4	1·00	331·0	212°·	308·7	1·00	295·0	212°·	576·5	1·00	275·0	212°·	851·0	1·00	270·8	212°·
¼	79·6	2·18	121·8	128·	142·6	2·16	121·2	133·	..	..	..	..	..	..	..	..
½	52·4	3·31	66·8	102·	93·	3·32	71·0	107·	174·3	3·308	74·0	110·	236·4	3·60	69·50	108·
1	33·54	5·02	32·05	82·8	56·	5·50	35·7	86·	101·9	5·657	38·94	89·	148·2	5·74	40·46	90·
2	21·95	7·90	13·98	70·9	34·7	8·9	16·6	73·	59·37	9·71	18·91	75·	77·9	10·92	18·60	75·
4	15·47	11·20	5·91	64·8	22·4	13·8	7·12	65·9	37·07	15·55	8·85	67·35	47·87	17·78	9·146	67·63
6	..	..	..	..	18·1	17·0	4·32	63·58	27·28	21·14	5·21	64·35	36·00	23·64	5·734	64·8

NOTE.—The Ratio in this Table gives the loss of heat by the naked pipe, the amount lost by a cased one being 1·0; thus a 4-inch naked pipe loses 8·9 times as much heat per foot run as it would lose if cased in 2 inches of felt, or 13·8 times as much as with 4 inches of felt, &c.

CHAPTER XIV.

ON THE TRANSMISSION OF HEAT BY CONDUCTION.

(310.) When heat is applied to the surface of a solid body, it is transmitted by conduction from particle to particle, and is gradually absorbed by the mass. The rapidity with which this proceeds, depends not only on the conducting power of the material, but also on its capacity for caloric, or specific heat, and its density. M. Cauchy has given a formula, by which the temperature attained at a given depth, in a given time, may be calculated. Table 87 gives the results by this formula as calculated by M. Péclet.

We may suppose the material to be a mass of infinite area and thickness, having thermometers imbedded in it at the given depths; the outer surface being suddenly heated to 1000°, the heat penetrates in one minute to the extent shown by the Table.

TABLE 87.—Of the DIFFUSION of HEAT by CONDUCTION, giving the Temperature attained in one Minute at different Depths in the mass of a body of infinite Size and Thickness, when the Surface is heated 1000°.

Nature of the Material.	Depth penetrated in inches.			
	$\frac{4}{100}$	$\frac{4}{10}$	4	40
	Temperature at that depth.			
Sand	988°	887°	154°	°·000
Limestone, used in building	997	973	745	·100
Iron	999	992	924	339
Marble, coarse-grained	999	993	938	439
Plaster, common	996	962	624	·000
Water, stagnant (heated at the top) ..	993	939	399	·000
Air ,, ,, ..	999	996	960	620

In the case of water and air, and indeed with liquids and gases in general, if heat be applied anywhere but at the top, currents are created by which the heat is rapidly diffused uniformly throughout the mass. For comparison with solids,

we must therefore heat these fluids at the top, in which case the particles first heated remain stagnant at the top and gradually transmit the heat downward from particle to particle by conduction only, as in the case of solids.

There is great uncertainty in applying these rules to air and gases, because of the great dilatation which heat occasions in those bodies; but it will serve to give some idea of the rapidity with which heat is transmitted. The Table shows that air even when stagnant transmits heat more rapidly than any other body; this is confirmed by experience: in the church of St. Roch the heated air after rising to the lofty roof passes off by crevices in the windows, &c., and it has been found that the temperature of the air at a height of 65 feet was never more than from ½ to 1½ degree higher than at 6 feet above the ground.

(311.) The Table seems to show that water carries heat downwards much more rapidly than we have supposed. This certainly is not confirmed by experiment, which has led to the conclusion that water is almost a perfect non-conductor downwards, as is stated in (108).

CHAPTER XV.

ON VENTILATION, ETC.

(312.) "*Air required for Respiration, &c.*"—In the act of respiration air becomes vitiated, the oxygen being transformed into carbonic acid. An ordinary man makes twenty respirations per minute, each of 40 cubic inches; therefore, admitting that air should not be respired a second time, we have $\frac{20 \times 40 \times 60}{1728}$ $= 28$ cubic feet of air thus vitiated per hour. But besides this a quantity of vapour is emitted, amounting, according to M. Dumas, to ·0836 lb. of water per hour, which would saturate (by col. 9 of Table 58).$\frac{\cdot 0836}{01179} = 7 \cdot 1$ lbs. of air at 62°, and if we allow that to be healthful and pleasant, the air should be half-saturated only, we require 14·2 lbs., or $\frac{14 \cdot 2}{\cdot 0761} = 187$ cubic feet of air (col. 3, Table 19); so that we have a total of $28 + 187 = 215$ cubic feet per hour, which happens to be the capacity of a cube 6 × 6 × 6 feet. This is the minimum quantity necessary for cleanly and healthy persons. For prisons, workhouses, &c., it should not be less than 350, and for hospitals 1000 cubic feet per hour per head.

(313.) It is also necessary to provide air to support the combustion of the lighting apparatus; a common candle requires about 11 cubic feet per hour, and an Argand lamp of large size about 45 cubic feet, assuming that one-third of the oxygen in the air is consumed.

(314.) "*Heat evolved by Respiration.*"—Another result of respiration is the emission of heat; for as we have seen in (23), combustion, by which heat is developed in our furnaces, consists in the combination of bodies (and mainly of carbon) with oxygen, so that respiration is a veritable act of combustion, and an amount of heat is evolved proportional to the carbon consumed. According to M. Dumas, the quantity of carbon given out by an ordinary man is ·022 lb. per hour, and by (24) the heat developed will be $12906 \times \cdot 022 = 284$ units per hour. A considerable part of this heat is absorbed by the vapour formed during respiration, and becomes latent, the amount from 62° being $\cdot 0836 \times (1178 - 62) = 93$ units, and the rest, or $284 - 93 = 191$ units, is dissipated by radiation to the surrounding objects and by contact of cold air. This amount of heat is sufficient to heat the air required for ventilation in most cases; if we assume the external air at 32° and the internal at 62°, or 30° difference, the heat emitted by one person, deducting the 93 units absorbed by the vapour, would suffice for $\dfrac{191}{30 \times \cdot 238 \times \cdot 0761} = 351$ cubic feet of air. We should require, however, another amount of heat to maintain the room at the required temperature, and for this, artificial heat would be necessary in most cases.

(315.) "*Ventilation.*"—We have now to consider the means by which the continuous renewal of the air in a room or building is to be effected. Let Fig. 93 be a room filled with heated air, and let the walls have the same temperature as the air; the temperature will not be the same throughout, but will vary slightly with the height, the top of the room being the hottest. If the walls are exposed to cooling influences, and become colder than the internal air, a down-current is established against the face of the wall, as shown by the arrows, Fig. 87, because the air in contact with the wall being cooled by it, becomes heavier than the air in the centre of the room, the former therefore descends, and the latter ascends, as shown by the arrows; and by the same means horizontal currents are established along the floor and ceiling, as shown by the figure.

(316.) If openings be made in the floor and ceiling, as at A and B, Fig. 88, the column A B would immediately ascend, being lighter than the column of cold external air with the same height, and the velocity of the motion can be easily calculated. Say the room is 10 feet high, with internal air at 62° and external air at 32°. We find by Table 19 that a column of air 10 feet high at 62° is equal to a column at 32° $\frac{10}{1\cdot061} = 9\cdot424$ feet high, we have therefore $10 - 9\cdot424 = \cdot576$ foot head to produce motion, and as we have two openings to deal with, we may suppose it to be divided between them, each taking $\frac{\cdot576}{2}$ ·288 foot head, and as in the case (145) we have $\sqrt{\cdot288} \times 8 = 4\cdot3$ feet per second, or with ·93 for co-efficient (148) $4\cdot3 \times \cdot93 = 4$ feet per second, and from this the area necessary can be easily calculated.

(317.) But it is obvious, that although a given amount of air might thus be caused to pass through a room, we should not have efficient ventilation, the portions of the room indicated by C and D would not be ventilated at all. If, as in Fig. 89, the openings were made at opposite ends of the room, the ventilation would not be much improved, the central part C receiving no benefit from the current of fresh air passing through the room.

(318.) If, as in Fig. 90, we admit the fresh air by very numerous holes, equally distributed all over the floor, and allow it to escape by similar holes in the roof, we then have perfect ventilation; but the plan is very difficult to carry out. Thus in Fig. 91 we have holes in the floor J and ceiling K, as in Fig. 90; but admitting the air by one large opening, A, and allowing it to depart by a similar opening at B, which is usually a practical necessity, the air takes the shortest course, and the holes in the direct line get the most of it, and it is exceedingly difficult to obtain uniformity in the ventilation.

(319.) There is another and a practical difficulty in this mode of ventilation. It is essential that the numerous inlet-openings should be of large area, otherwise the velocity of the air would be so great as to be a nuisance, especially for summer ventilation, when cold air is admitted; but if the velocity be very low, it is apt to be greatly disturbed, and the current even reversed by light winds in certain directions, so that it is essential that the velocity through the inlet and outlet openings should be considerable.

(320.) In Fig. 92 we have a room ventilated on a totally different principle. The heated air enters by one large opening at A, and mounts directly to the ceiling. The cooled air is drawn off by an opening, B, leading to an air-chimney, C, in which the air is maintained at all seasons at a high temperature, thus causing a rapid current, too strong to be seriously affected by winds. The heated air entering the room rises in a body to the roof, and is distributed in horizontal strata, having the same temperature at the same level all over the room, as in Fig. 93. This is a natural circumstance where there are no disturbing causes, and an important one for our purpose. Let Fig. 94 be a room like Fig. 93, but say that by some accidental cause the air at one end of the room is more heated than at the other, the beds of air will in this case be at an angle with the horizon, as per Figure. It will be seen that the *mean* temperature of the air at one end of the room is 10° higher than at the other, it will therefore weigh less, and motion will ensue in the direction of the arrows until uniform horizontal temperatures are obtained, as in Fig. 93. By the action of the ventilating-shaft, the air is regularly drawn down, and the number and position of the exit-orifices (on the plan) has no influence on the ventilation. But if, as in summer, the air admitted at A is cooler than that in the room, this arrangement of the openings would be the worst possible; for, as in Fig. 95, the cold air would remain at the bottom, passing away along the floor to the opening B without ventilating the room at all. We require in this case an opening at the top of the room, or at least above the heads of the people, as at D. The cold air admitted at B will then spread horizontally all over the floor, and ascending regularly by the action of the chimney and by the heat imparted to it will depart vitiated by the opening D. If instead of admitting the cold air at A, we let it in by an opening in the ceiling at E, the cold air would fall direct to the floor because of its superior weight, and would then be distributed and rise uniformly to the opening D as before; but the descent of the cold air would be felt as a draught by those in and near its course.

(321.) "*Summer and Winter Ventilation.*"—It will be seen from this, that the arrangement of the orifices of access and exit of air must be varied with the seasons. In winter, when the air entering is previously heated, and is of higher temperature than the room, it may enter by one or more openings in any part of the room, and should be drawn off by one or more openings near the ground into the air-chimney. In summer the cool external air should be admitted by openings in the floor, and should pass into the chimney by openings near the top of the room, or at least above the heads of the inmates. For both cases we have supposed a good air-chimney to be used, and an active draught to be maintained, summer and winter, when ventilation is required.

(322.) With *natural ventilation*, as it is termed, as per Figs. 88–91, no change is necessary with the seasons. In all cases the air should enter by numerous openings in the floor, and depart by similar openings in the ceiling. No chimney is necessary, but the exit-opening should be provided with a hood to prevent the action of the wind disturbing the draught. This mode *appears* by far the cheapest and simplest; but practically it is very uncertain and frequently very ineffective. It is almost the only plan used in our churches and chapels, and by universal experience is far from satisfactory, especially in summer, although supplemented by opening windows and other highly objectionable means. It is deeply to be regretted that the acknowledged difficulties of accomplishing effective ventilation have led to the whole question being virtually abandoned by architects and others designing our public and private buildings.

TABLE 88.—Of the PROPORTIONS of VENTILATING FANS.

Diameter of Fan in feet.	Size of Exit opening in feet.			Revolutions per Minute.	Cubic feet of Air per Minute.	No. of People.	Approximate weight for 30 feet fall, in cwts.
	Width.	Length.	Area.				
3	1·5	1·8	2·7	83	810	216	6·5
4	2·0	2·4	4·8	62	1440	384	11·5
5	2·5	3·0	7·5	50	2250	600	18
6	3·0	3·6	10·8	42	3240	864	26
7	3·5	4·2	14·7	36	4410	1176	35
8	4·0	4·8	19·2	31	5760	1536	46
9	4·5	5·4	24·3	28	7290	1944	58
10	5·0	6·0	30·0	25	9000	2400	72

(323.) "*Mechanical Ventilation.*"—Ventilation may also be very effectively, and in many cases cheaply accomplished by *a fan* worked by manual or engine power. In large establishments where an engine is used for other purposes, a large fan, driven at a low speed, may be used; in other cases a fan worked by a man will answer the purpose. The great objection to this mode is, that of necessity the ventilation continues only so long as the machine is in motion; and for such cases as hospitals, &c., where ventilation is required without any intermission night and day, this plan is hardly admissible (363). The use of an air-chimney is in such cases much preferable. The fire can be be arranged to burn for some hours; and the mass of brickwork in the chimney having been well heated during the day, retains so much heat as to keep up the ventilation pretty well until the morning, when the fire is relighted (365.)

(324.) There are two kinds of fan which may be used for ventilating purposes, one compressing and the other exhausting or rarefying the air operated on. The former is preferable in most cases; the latter having the objection that it causes draughts of cold air to enter by all crevices in the floors, windows, and open doors, &c. It is evident from (321) that a compressing-fan should deliver the air at the floor-line, the orifices of discharge being also near the floor-line in winter, and near the ceiling in summer.

(325.) The application of the rules and principles we have arrived at in the preceding investigation will be best understood by examples. We will take three cases: the first being a school, in which the air is heated and also *forced* into the room by stoves; the second, a chapel, heated by ordinary hot-water pipes and ventilated by a compressing-fan, driven by a weight wound up by manual labour during the preceding week; and the third, a hospital, heated by hot-air pipes and ventilated by a draught-chimney with a fire maintained in it at all seasons.

(326.) "*Schools, &c.*"—Figs. 96–98 show a school for 100 boys, on the plan proposed by Péclet, heated and ventilated by a stove; this is constructed with a double case, the inner one containing the fire and the smoke, &c., which passes by a long pipe to a chimney, N, at the end of the room; the air to be heated passes between the two cases, being drawn from the room itself when the fires are first lighted by closing the damper B and opening the door at A; but when *ventilation* is required during school hours, the damper B is opened wide, and A is closed, the air being drawn through a grating at C by the channel D from the exterior. The air between the cases being highly heated, we have a column the height of the stove dilated to say $\frac{1}{3}$rd the volume of the air in the room, and a powerful current is created, by which air is *forced* into the room, passing out by openings at the top of the case. The air *in the room* is further heated by the flue-pipes, and in both cases it will rise to the ceiling by reason of its levity, and would pass away if openings were made at the top of the room, and its heat would be wasted; but openings for ventilation are made about 18 inches or 2 feet from the ground at E F, which can be opened and closed at pleasure. The air is therefore distributed all over the room in horizontal layers, having the same temperature throughout (320), and is regularly drawn down, becoming cooler as it descends by giving out its heat to the walls. The walls are also further heated by *radiation* from the stove and pipes.

(327.) For summer ventilation when of course no fire is needed in the stove, a small special fire is maintained at the base of the chimney N by a furnace H, the openings E and F are closed for reasons given in (320), and a large register opening, G, is opened by the cords J J, this opening may be as near the top of the room as convenient; it must be above the heads of the inmates, as we have seen in (320).

The only objection to placing it quite at the top of the room, is that the effective height of the chimney and the power of the draught are reduced thereby, the *effective* height in fact is the distance G L, which of course is reduced by placing G at a higher level.

(328). In the cold season, the fire must be lighted an hour or two before school hours, in order to warm the walls and air in the room, which indeed is the principal object of the heating apparatus; for, as we have seen (314), each individual emits heat enough to warm the heat required for ventilation, and the fire may die out an hour or so before the time of dismission, the walls, &c., containing a volume of heat that is not quickly dissipated. When first lighted, *all* the exit openings, E F G, as also the furnace door at H and its register K, are closed, the stove heating the air in the room only, and this enters by the door A, the damper B being closed. When the pupils have assembled, B E and F are opened, and A is closed, &c., &c. The fire is regulated by a damper at M.

(329.) In summer the fire need not be lighted in H before the hour of assembling, K is opened only sufficiently to maintain the fire in a fair state of combustion, A E and F are closed, and B and G opened.

(330.) We may calculate the proportion of the entire apparatus by the rules already given.

"*Heat dissipated by the Walls, &c.*"—The building exposes an area of 210 square feet in windows, and 1340 square feet of 14-inch walls; assuming for the air, an internal temperature of 60° and external 30°, the loss by the windows will be 12·3 units per square foot (294), and in our case 12·3 × 210 = 2583 units per hour, and the walls (Table 78) will lose 1340 × ·159 × 30° = 6400 units, giving a total of 8983 units per hour lost by the building after it has been heated to the standard temperature. But a great amount of heat must be absorbed by the walls before they can be brought up to that standard; by (282) and Fig. 80, we see that with 14-inch walls, and air at 60° and 30°, the *mean* temperature of the wall is $\frac{48^\circ\cdot 02 + 34^\circ\cdot 2}{2} = 41^\circ\cdot 11$, or say 41°; they have therefore to be heated in the morning from 30° to 41°, or 11°, and as they contain about 1570 cubic feet, weighing by Table 46, = 115 lbs. per cubic foot, and the specific heat of brickwork (burnt clay) being ·185 by Table 1, they will require 1570 × 11 × ·185 = 367420 units of heat to raise their temperature from 30° to 41°. In our case they will receive it from two sources, from the heated air in the room and by direct radiation from the stove and stove-pipe. The amount that can be received by contact of air is at first, when the walls are cold, or at 30°, by (272) and Table 74 = ·435 × 30 = 13·05 units per square foot; but at the end of the operation, when the walls are heated to their standard internal temperature of 48°, we have only ·435 × (60 − 48) = 5·22 units; the mean is $\frac{13\cdot 05 + 5\cdot 22}{2}$ = 9·14 units per square foot per hour.

(331.) "*Time required to heat the Walls.*"—At low temperatures, heated bodies give out their heat by radiation and contact in nearly equal proportions, even so high as 210°, the Table in (302) shows that a 3-inch pipe loses 157·5 units by radiation, and 145·5 by contact of air. But at high temperatures the ratio is very different, as is shown by (246), where a 3-inch stove-pipe at 800° loses ·7 × 7 = 4·9 units per degree by radiation, and ·6256 × 2·2 = 1·38 unit only by contact of air. While the walls are receiving a *variable* quantity of heat by contact, the heat received by radiation will be nearly *constant*, because the slight variation in the temperature of the walls has but little effect with the great difference that exists at high temperatures; thus, at first the difference is 800° − 30 = 770°, and at last 800° − 48° = 752°, so that it is practically constant.

Admitting the ratio to be as we found it above, 4·9 to 1·38, we have $\frac{13 \cdot 05 \times 4 \cdot 9}{1 \cdot 38} = 64$ units by radiation, which is constant, and added to the *mean* received by contact, makes $64 + 9 \cdot 14 = 73 \cdot 14$ units per square foot, and the walls will receive $73 \cdot 14 \times 1340 = 97307$ units per hour. But while this is going on, heat is being dissipated by the external surface of the walls and the windows; this is 0 at first, the walls, &c., having the same temperature as the external air, but when the standard temperature is attained, it becomes, as we have seen, 8983 units per hour, the mean is $\frac{8983 + 0}{2} = 4492$ units, and hence the amount retained by the walls is $97307 - 4492 = 92815$ units per hour, and to bring them up to the standard temperature, we shall require $\frac{367420}{92815} = 4$ hours nearly.

The *maximum* quantity of heat received by the walls is $64 + 13 = 77$ units per square foot, or $77 \times 1340 = 103180$ units per hour. A pound of coals contains about 12000 units (90), but of this, about 3546 units are lost by the chimney, see (243), and the useful effect is reduced to $12000 - 3546 = 8454$ units: we shall therefore require $\frac{103180}{8454} = 12$ lbs. of coals per hour.

With sheet-iron flue-pipes (243) we require 4·4 square feet of surface per lb. of coals, in our case $4 \cdot 4 \times 12 = 53$ square feet. The flue-pipe in Fig. 96 is 42 feet long, and with 5 inches diameter the area will be $\frac{3 \cdot 14 \times 5}{12} \times 42 = 55$ square feet.

(332.) "*Time to cool down the Building.*"—The time required to cool the building down to the external temperature is much greater than that required to heat it. The loss at first is 8983 units, gradually reducing to 0, the mean being 4492 units per hour, and the building will cool down in $\frac{367420}{4492} = 82$ hours. In most cases about sixteen hours only elapse before the fire is relighted, so that what really happens is that the walls retain a large portion of their heat continually, and require less time to raise them to the standard temperature.

(333.) To maintain the heat and ventilation when once the standard was attained, would require very little heat. Allowing 220 cubic feet of air per head, or $220 \times {\cdot}076 = 16 \cdot 72$ lbs., we should require for 30° increase, $16 \cdot 72 \times 100 \times 30 \times {\cdot}238 =$

11938 units per hour, which added to that dissipated by the building, makes a total of 11938 + 8983 = 20921 units per hour. But we have seen by (314) that each individual emits 191 units, and in our case we have 191 × 100 = 19100 units of heat from them, or nearly the amount required for ventilation and for the walls and windows: this agrees with our experience that in a crowded room, artificial heat is not necessary, except to warm the walls, &c., beforehand, and in most cases, the proportions of the heating apparatus must be fixed with special reference to the preliminary heating of the building, which we have done in this case.

(334.) It is important to remark, that the time required to heat the building is much less with stoves, &c., whose temperature is very high, than with steam or hot-water pipes whose temperature is low, and there is a special reason for this. The walls *cannot* absorb more than 13·05 units per square foot from the air with the temperature we have assumed, whatever may be the power of the apparatus, the temperature of the internal air being limited to 60°; with ordinary steam-pipes, as we have seen by (302) and (331), the heat emitted by radiation is about equal to that emitted by contact, and the total is 26 units. This is the maximum when the walls are cold; at the end of the operation it is 5·22 units by contact of air (330), and say 5·22 more by radiation, or 10·44 units from both sources, the mean is therefore $\frac{26 + 10 \cdot 44}{2} = 18 \cdot 22$ units, and the walls will receive $18 \cdot 22 \times 1340 = 24415$ units per hour. The walls and windows will dissipate 4492 units per hour on external objects as before, hence the walls retain 24415 − 4492 = 19923 units, and the time to attain the standard temperature would be $\frac{367420}{19923} = 18$ hours with steam-pipes or hot-water pipes at the ordinary temperature of 210°, whereas with a stove, the time can be reduced to four hours, as we have shown (331).

(335.) "*Area of Inlet and Outlet Openings, &c.*"—The area of the openings must be calculated for summer rather than for winter ventilation, because the heat required to produce the former has to be obtained by a special fire maintained for no other purpose; the less the air can be heated, the less costly is the operation, but with low temperatures large apertures are necessary.

Let us assume in our case that the air in the chimney N is heated 50° above the temperature of the room and external air, which may both have a summer temperature of say 70°, the air in the chimney will therefore be at 120°, and the *effective* height G L being about 12 feet, this will be equal by Table 19 to $\frac{12 \times 1\cdot082}{1\cdot184} = 10\cdot97$ feet of exterior air, and giving $12 - 10\cdot97 = 1\cdot03$ feet for generating velocity (145). As this has to be given at two places (inlet and outlet), we must divide the available head into two equal portions, allowing $\frac{1\cdot03}{2} = \cdot515$ foot for each, and the theoretical velocity will be $\sqrt{\cdot515} \times 8 = 5\cdot74$ feet per second, but with an orifice in a thin plate we have seen (148) that the real discharge is ·65 of the theoretical, and we have $5\cdot74 \times \cdot65 = 3\cdot73$ feet per second, and as we have $\frac{220 \times 100}{3600} = 6\cdot11$ cubic feet of air per second, the openings for summer ventilation must have an area of $\frac{6\cdot11}{3\cdot73} = 1\cdot64$ square foot. The circular inlet tunnel D must therefore be 18 inches diameter, and the register G, which exposes rather less than half its total area as available openings (say ·4), must have a *total* area of $\frac{1\cdot64}{\cdot4} = 4\cdot1$ square feet, or say 2 feet 3 inches diameter.

(336.) The quantity of fuel required in the furnace at H can be easily calculated, it being the amount necessary to heat $16\cdot72 \times 100 = 1672$ lbs. of air 50°, or $1672 \times 50 \times \cdot238 = 19{,}697$ units, and if we allow that in such a case only half the total heat in coals is available, the rest being dissipated by radiation, &c., we shall require $\frac{19{,}697}{6000} = 3\cdot28$ lbs. of coal per hour. In order to obtain sufficient inlet area at the stove for summer ventilation, the damper M should be closed and the door A as well as the furnace door may be thrown open or removed for the season. It will be seen from this, that we require about $\frac{1}{4}$ as much fuel to effect the summer ventilation as to heat and ventilate the room in winter.

These particulars of apparatus, applicable to a school for 100 boys, may be easily applied by proportion for fixing the sizes, &c., in other cases.

(337.) "*Chapels, &c.*"—The case of a church or chapel, used only one day per week is peculiar; the whole building exposed for six days to cooling influences, will have taken pretty nearly the temperature of the atmosphere, and will require a large amount of heat to bring its *walls*, &c., up to the standard temperature desired. The case of a church of large dimensions is investigated in the next chapter, see (366, &c.); we will now consider that of a chapel to hold 400 persons, having the sizes and form shown by Figs. 103 to 108.

"*Heating Apparatus.*"—The heating apparatus has to do two things: it has first to heat the walls of the building before it has to be occupied; and it has to heat the air entering the building to supply 400 people during the hours of public worship. It is necessary to determine which of these two is the greater, and to fix the size of the apparatus for that amount. Assuming that the external air is 30° and the internal 60°, the conditions of the building are similar to those in Fig. 80, the building being exposed on all sides to cooling influences.

Allowing 220 cubic feet of air per head per hour as in (312), we shall have $220 \times 400 = 88{,}000$ cubic feet, or $88{,}000 \times \cdot 0761 = 6697$ lbs. of air per hour, to heat which from 30° to 60° will require $6697 \times 30 \times \cdot 238 = 47816$ units of heat.

(338.) The walls being 30° at first and the internal air 60°, taking the value of A from Table 74 at ·4133, will absorb $\cdot 4133 \times 30 = 12 \cdot 4$ units per square foot. The area of the walls is about 3258 feet inside and 3368 feet outside (deducting the windows and doors), taking the mean, which is near enough for our purpose, we have 3313 square feet, which will absorb $3313 \times 12 \cdot 4 = 41081$ units per hour. Then the windows, say 20 in number, each $6 \times 4 = 480$ square feet, by (294) will lose $12 \cdot 3$ units per square foot, or $480 \times 12 \cdot 3 = 5904$ units per hour, making together $41081 + 5904 = 46985$ units, or very nearly the same as for the air to keep up ventilation, which we found to be 47816 units.

(339.) By Table 66 an *enclosed* pipe 3 inches diameter, heated to 200° with air at 60°, gives 184 units per foot run, but with air at 30°, as in our case, this will be increased to $\frac{184 \times (200 - 30^\circ)}{200 - 60^\circ} = 223$ units per foot, and taking the maximum quantity or that required by the air, we shall require $\frac{47816}{223} = 214$ feet of 3-inch pipe, which may be arranged in two lines as in Fig. 104. These pipes are carried by rollers, like Fig. 53, which are supported on cross-beams, H, Fig. 107, built into the brick side-walls of the channels in which the pipes are enclosed.

The boiler may be of the common horse-shoe form, and should be placed below the level of the chapel floor so as to obtain a simple circulation (258); where this is quite impracticable it is *possible* to work a coil of pipes by a boiler placed above it, as in (265); in that case the feed-cistern must be placed in the roof at A, or perhaps at B in Fig. 106; but this plan should never be adopted if it can possibly be avoided. For the maximum of 47816 units we should require about $\frac{47816}{4000} =$ 12 lbs. of coal per hour, and a square foot of fire-grate (120). With a very short and small boiler, such as is commonly used for such purposes, we should not obtain usefully much more than the radiant heat (88) in the fuel, and much of that will be lost, so that we have taken in the above the low economic value of 4000 units per pound of coal.

(340.) "*Time required to heat the Apparatus itself, &c.*"—The water-pipes, boiler, &c., will never be cooled down to 30°, the external temperature. We have seen in (22), that at a depth of about 20 feet we come to a stratum having constantly the mean temperature of the year at that place, and when the earth is covered by a building and protected from the cooling action of the atmosphere, the ground will have a temperature varying very little from the same temperature (279). By Table 26 the mean temperature of the year at London is 50°·7, and we may take the minimum temperature of the pipes, &c., at 50°. The boiler and pipes contain about 3700 lbs. of cast-iron and 1100 lbs. of water; to raise the water 160°, or from 50° to 210°, will require $1100 \times 160 = 176000$ units. Cast-iron having by Table 1 a specific heat of ·13, we shall require $3700 \times 160 \times \cdot 13 = 76960$ units for the boiler and pipes. But while heat is being received from the fuel, a considerable amount will be given out to the air; at first this will be 0 and at last 47816 units, the mean is therefore $\frac{47816 + 0}{2} = 23908$ units per hour; while 47816 units are received, 23908 units are parted with, and 23908 units are retained, and to raise the whole apparatus to its maximum temperature of 210° we shall require $\frac{176000 + 76960}{23908} = 10 \cdot 6$ hours; but during that time it will give out to the air and the walls, &c., half its maximum amount of heat. This time might be reduced probably to nearly one-half by forcing the fire.

(341.) "*Time required for heating the Walls, &c.*"—By Fig. 80 it will be seen, that when the standard temperature is attained, the internal surface of the wall will be 48° and the external surface at 34°; the mean temperature of the wall will therefore be $\frac{48 + 34}{2} = 41°$, or 11° higher than the external air. The building contains about 451000 lbs. of brickwork, the specific heat of which being by Table 1 ·185, we shall require $451000 \times 11 \times \cdot185 = 917780$ units to bring them to the standard temperature. When that temperature is attained, the difference between the air and the walls will be $60 - 48 = 12°$, and the walls will receive $\cdot4133 \times 12 = 4\cdot96$ units per square foot per hour, and the same amount will be dissipated by the external surface. But at first the walls will receive, as we have seen (338), 12·4 units and the external surface will dissipate 0. The *mean* heat received is therefore $\frac{12\cdot4 + 4\cdot96}{2} = 8\cdot68$ units, and the mean heat dissipated is $\frac{4\cdot96 + 0}{2} = 2\cdot5$ units, so that $8\cdot68 - 2\cdot5 = 6\cdot18$ units is retained by the wall, or $3313 \times 6\cdot18 = 20474$ units per hour by the whole surface of the walls, and to obtain the standard temperature we shall require $\frac{917780}{20474} = 44\cdot8$ hours, the heating apparatus giving its full maximum effect all the time. During the time that the apparatus itself is being heated, we found that it gave out to the building only half its maximum effect, so that the 10·6 hours at half power is equivalent nearly to 5·3 hours of full power, and 5·3 hours goes for nothing, and the time required from lighting the fire is $44\cdot8 + 5\cdot3 = 50\cdot1$ hours.

(342.) "*Time required to cool the Walls, &c., to the External Temperature.*"—When the fire is put out, the building will gradually cool down to the temperature of the external air. At first it will lose 4·96 units per square foot per hour, and at last 0, the mean loss is $\frac{4\cdot96 + 0}{2} = 2\cdot5$ units, and for the whole surface of walls $3313 \times 2\cdot5 = 8282$ units per hour. To this has to be added the heat lost by the windows, which at first is 12·3 units, and at last 0, the mean being $\frac{12\cdot3 + 0}{2} = 6\cdot2$ units per foot, or $480 \times 6\cdot2 = 2976$ units per hour. We found by (340) that the water in the apparatus required 176000 units, and the iron in the pipes, &c., 76960 units to heat it, and these will of course give out the same amount in cooling down. The walls will give out 917780 units as in (341), so that the time to cool down is $\frac{176000 + 76960 + 917780}{8282 + 2976} = 104$ hours, or 4·3 days.

(343.) "*Heat required to keep up the Temperature of the Building.*"—When the building is once raised to its standard temperature, a much smaller amount of heat per hour will suffice to maintain it. The walls receive, transmit, and dissipate on external objects 4·96 units per foot, as we have seen (341), and the total loss by the walls is $3313 \times 4 \cdot 96 = 16432$ units per hour. The windows lose $480 \times 12 \cdot 3 = 5904$ units, making from both sources $16432 + 5904 = 22336$, requiring $\frac{22336}{4000} =$ 5·6 lbs. of coal instead of 12 lbs., the amount used in getting up the temperature. During the hours of worship, a large amount of heat is carried off by the air required by the congregation, which is forced through the building by a fan and passes away highly heated into the atmosphere, but this loss of heat is more than supplied by the animal heat emitted, which being by (314) 191 units per head, we shall have from this source $191 \times 400 = 76400$ units, or nearly 3½ times the amount required to keep up the temperature of the walls, &c., and $\frac{76400}{47816} = 1 \cdot 6$ times the maximum power of the heating apparatus.

(344.) The whole week of 168 hours may be divided into three portions: 50 hours being spent in getting up the temperature, 14 hours in maintaining it during public worship, and 104 hours in cooling down again. The weekly consumption of coals would be $(50 \times 12) + (14 \times 5 \cdot 6) = 678$ lbs. or 6 cwt.; if the temperature were maintained throughout the week by continuous slow firing, the consumption would be $168 \times 5 \cdot 6 =$ 940 lbs. or 8·4 cwt. With regular and slow firing the economy of fuel would be considerable, and the actual consumption would probably be not more than 7½ cwt., or 1½ cwt. more than by intermittent firing; the chapel would always be ready for week-night and occasional services, and the conveniences of this mode of heating are so great that it should become general.

(345.) "*Ventilation.*"—Allowing 220 cubic feet of air per head per hour, we require in our case $\frac{220 \times 400}{60} = 1467$ cubic feet, or

$1467 \times \cdot 0761 = 112$ lbs. of air per minute; and this quantity has in our case to be supplied by a fan, the construction and details of which are given by Figs. 63 to 66. This fan is driven by a weight, which may be wound up by manual power during the week, being worked only during the four hours of worship, &c. The power which such a fan will take depends greatly on the velocity of the current of air issuing from it: in fact the power is directly as the square of the velocity, omitting the consideration of friction of the machine. There is a practical limit, however, to the extent to which the power may be reduced by reducing the speed; for at very low velocities, the current is apt to be retarded, and even overpowered and reversed, by the wind when in certain directions. Allowing 5 feet per second as a good velocity, and admitting as in (362) that the real velocity is $\cdot 423$ of the theoretical, we have the latter $\frac{5}{\cdot 423} = 11 \cdot 8$ feet per second, and the head due thereto by the laws of falling bodies is $\left(\frac{11 \cdot 8}{8}\right)^2 = 2 \cdot 17$ feet, and the mechanical work to be done is to raise 112 lbs. of air $2 \cdot 17$ feet per minute, or $112 \times 2 \cdot 17 = 243$ foot-pounds, and for four hours the duration of the services—this is equal to $243 \times 4 \times 60 = 58320$ foot-pounds, and if the weight could be raised 30 feet, we should require $\frac{58320}{30} = 1944$ lbs., or say 18 cwt. in round numbers. An ordinary man can easily raise 3000 foot-pounds per minute by a winch: he could therefore raise the weight in $\frac{1944 \times 30}{3000} = 19$ minutes or rather more, and of course this could be done at any time during the week. With large fans, the weight becomes very great, and the labour excessive.

(346.) Figs. 63 to 66 give all the necessary details of the fan and the gearing for working it. One of the principal things to be attended to is to make all the parts *very light*, and another is to avoid enlargements of the area in the air-passages, by which velocity is lost after having been got up elsewhere. The air-passages should either have the same area throughout or, what is perhaps better still, a diminishing area, so that the velocity is gradually got up from point to point, but never suffered to fall again by an enlargement of channel: see (161). In Fig. 63, A is the fan, 5 feet diameter, having six blades of light sheet zinc or iron, which are curved to a radius of $\frac{3}{8}$ths of the diameter, or in our case 22½ inches, the describing circle or centre being $\frac{1}{3}$rd of the diameter, or 20 inches, as in the Figure; we thus obtain air-passages of 7 inches at the smallest part between blade and blade, increasing to the circumference. The blades are supported by two sets of light deal arms, fixed to light cast-iron centre-bosses, keyed on the wrought-iron shaft. The centre openings are half the diameter of the fan, or 30 inches, and the distance between the side cheeks is the same dimension. The area of the two central 30-inch openings is 9·8 feet; the area of the passages between the fan-blades, at the narrowest part is $\frac{7 \times 30 \times 6}{144}$ 8·75 feet; and the outlet-opening is $3 \times 2\cdot5 = 7\cdot5$ square feet: the velocity is thus gradually got up. The case may be made of light deal or any other convenient material. Table 88 gives the general proportions for various sizes.

The gearing should be very light in pitch, and truly fitted in the teeth. The pinion B is made to slide on a feather let into the fan-shaft, so as to draw endwise out of gear with its wheel when the weight is being wound up, which is done by a handle, C, on the second-motion shaft. This handle should be removed after the weight is wound up; and the machine is stopped till required by the break D, which is kept in contact with the unturned wheel F by the weight G. To start the machine, the break-lever is raised by its handle H, and is kept up by the spring holder J. The lever D is fitted with a block of hard wood, curved to fit the wheel F. The frames are secured to the floor by bolts in their feet, and to one another by a cross-stay at K. The barrel carries a light wire or hempen rope, which is led under a large guide-pulley, and thence over another at the top of the building, so as to raise the weight in a channel provided for it in the wall, being guided and kept in position there by long vertical guides of iron or hard wood.

(347.) The centre of effort of a fan is about $\frac{7}{8}$ths of the extreme diameter, or in our case $\frac{5 \times 7}{8} = 4\cdot35$ feet, the circumference of which being $4\cdot35 \times 3\cdot14 = 13\cdot7$ feet, we shall require for 11·8 feet per second, or 700 feet per minute, $\frac{300}{13\cdot7} = 50$ revolutions per minute. With so slow a speed there would be no objectionable noise with well-fitted wheels; the pitch should be fine and the requisite strength obtained by width on the face. The strength and sizes of the wheels must vary with the weight to be carried (see Table 88); the style of workmanship should be like that of a church clock, &c.

(348.) "*Air-passages, &c.*"—The air from the fan descends below the floor of the boiler-house, and branching right and left, enters the channels in which the hot-water pipes are fixed, A, B, C, Fig. 104, from which it is conveyed by branch channels D, E, &c., into four other channels, G, H, J, K, which deliver the air by apertures under the seats in each pew, as at L, M N, O, &c. The arrangement of these openings is a matter of importance to avoid objectionable draughts. Figs. 101 and 102 illustrate two modes of doing it, which may be varied to suit local circumstances, &c. The plan 102 is much the best.

To supply air to the galleries, channels are taken from G and K, as at P, P, P, &c., and are continued by other channels, R, R, &c., in the walls, and thence into the pews by branches S, S, S, &c. To prevent the accumulation of heated air under the galleries, short channels are made at intervals, as at T, T, T, &c., covered with ornamental open gratings. The heated air passes away out of the building by channels W, W, &c., which deliver into the roof, where it finally escapes by the cowl V. In winter the heated air escapes by openings at Y. The reason for placing it at so low a level is that the heat may be retained as long as possible, for the purpose of heating the building. In summer the air may escape by openings near the ceiling at X. These openings, Y and X, must be closed by sliding registers, which should be connected together, so that when one is open the other is closed, &c. For summer ventilation the openings might be made through the ceiling at Z Z, which must be closed in winter. For the inlet of air to the fan a permanent provision should be made by a louvréd window, U, having a *free* area of about 10 or 12 square feet, and well-fitted shutters should be provided to close it completely while the walls are being heated (341.)

(349.) The area of the channels should be proportional to the volume of air passing through them. With a velocity of 5 feet per second, or 300 feet per minute, allowing 220 feet of air per head, each of the main air-channels as they leave the fan should have an area of $\frac{220 \times 400}{2 \times 60 \times 300} = 2 \cdot 5$ square feet. A section of these channels is given in Fig. 107. The pipes are carried by rollers on cross-timbers, H, built at intervals across the channel, and they should coincide with the branch-channels

D, so that the contraction occurs at one place; and then to obviate it and preserve the area, the bottom should dip in a curve, as in Fig. 108. The hot-water pipes are enclosed completely in a wooden casing; the air is admitted by a row of holes, say 1¼ inch diameter, whose total area must be equal to the area of the two main channels, or 5 square feet, or $144 \times 5 = 720$ square inches. With ·8 for coefficient of contraction (148), the area of 1¼ is $1\cdot22 \times \cdot8 = \cdot976$, and we shall require $\frac{720}{\cdot976} = 740$ holes, which distributed on a length of 214 feet or 2568 inches, must be $\frac{2568}{740} = 3\frac{1}{2}$ inches, centre to centre.

(350.) In fixing the area of the branch-channels, we may allow $\frac{720}{400} = 1\cdot8$, say 2 square inches per head. Each pew for four persons requires $2 \times 4 = 8$ square inches, and there would be no loss of effect here if it were made much larger, say double. Loss of power arises from loss of velocity only where that velocity has to be regained, which is not the case when the air is finally discharged (161). The galleries may contain 90 people, and the 8 channels, P, R, and S, supplying each 11 people, must have an area of $11 \times 2 = 22$ square inches; the 7 channels, D D, have to supply 110 people, or 16 each, including P, R, and S, and must be $16 \times 2 = 32$ inches area; the others, E, E, have only eight to supply, and must be $8 \times 2 = 16$ inches area.

(351.) The eight exit openings Y and channel W should have an area of at least $\frac{720}{8} = 90$ square inches; but here also there will be no loss by using a larger area. Allowance should also be made for the obstructions of the grating, and the *contraction* (148) which a fluid experiences in passing by a thin-lipped orifice. With ·65 for coefficient, each opening must have a *free* area of $\frac{90}{\cdot65} = 140$ square inches; the bars of the grating may probably occupy ⅕th of the total area, which should therefore be $\frac{140 \times 5}{4} = 175$ square inches, say 15×12 inches.

(352.) "*Hospitals.*"—The ventilation of a hospital must be more perfect, powerful, and uniform than any other; the state of health of the inmates necessitates a larger volume of fresh air than is necessary for persons in good health, and the ventilation must be continuous night and day without intermission.

For perfect comfort, the *walls* should be at least as hot as the air in the room, which as we have seen (297) is impossible, where they have to be heated by the air in the room. Mechanical ventilation by a fan is not admissible, because in most cases hospitals are too large to be thus ventilated without an engine, to work which night and day is expensive; besides, as the work to be done cannot be remitted even for an hour, we should require a duplicate set of apparatus in case of repairs. A chimney in which a draught is maintained by a fire is the best plan, because the mass of brickwork in the chimney retains so much heat as to maintain a fair draught for hours after the fire has been suffered to die out: see (365).

(353.) Figs. 109–112 represent a small hospital, or the wing of a large one, in which heated air is supplied in winter by a cockle or hot-air stove, and the ventilation is maintained at all seasons by a draught chimney. A is the hot-air apparatus, consisting of a collection of pipes, B, open at both ends and built into the side-walls, which are retained in position by clamp plates and bolts, the fire from the furnace at C circulates among the pipes in its passage to the chimney D, which serves for the escape of the smoke, &c., as well as the foul air from the rooms. The air to be heated enters the pipes at E, and passes through them into the chamber F, from which it proceeds by the channels G G, which run longitudinally the whole length of the building. Other channels, H H, are made in the main walls by which the highly-heated air passes to the top of the building, thence descending by the channels J J to the bottom again, imparting the requisite heat to the walls in its passage. From the channels J J, branch-pipes, N N, are laid, discharging the heated air (which by this time is cooled down to the proper temperature) into other long channels, O O, and thence into the room by openings under each bed and at suitable places in the offices, &c., &c., in the ground floor.

(354.) The air thus received into the rooms passes through and ventilates them, escaping by orifices, K K, &c., in the ceiling, into foul air channels, L L, which conduct it to the end of the building, where it enters the descending shafts, M M, which communicate with the chimney D, by which it is finally discharged into the atmosphere.

The heated air enters the walls at a much higher temperature than is desirable for the rooms, but the walls absorb its surplus heat, and become heated perfectly throughout their mass, the air having for the most part to traverse the height of the building twice, before it escapes into the rooms. The direction of the currents is shown by arrows, and it will be observed that the vitiated air moves off directly into the atmosphere, without mixing with the rest of the air in the room.

(355.) "*Loss of Heat by the Building, &c.*"—Allowing 1000 cubic feet of air per head per hour, and that we have 50 × 3 = 150 inmates, we shall require 150000 cubic feet of air at 60°, or ·076 × 150000 = 11400 lbs. of air per hour. If we assume that the interior surface of the walls is at 60°, with a thickness of 2 ft. 3 in., we shall have (288) by the formula $U = \dfrac{Q \times (t - T')}{1 + Q\dfrac{E}{C}}$

or in our case $\dfrac{1{\cdot}134 \times (60 - 30)}{1 + \left(1{\cdot}134) \times \dfrac{27}{4{\cdot}83}\right)} = 4{\cdot}63$ units per square foot per hour, and as we have 11790 square feet of wall surface (windows excepted), the heat dissipated by them will be 11790 × 4·63 × 54567 units of heat per hour. This heat has to be supplied by the 11400 lbs. of air entering the building, and to do that, it must be cooled $\dfrac{54567}{11400 \times {\cdot}238} = 20°$, and as it leaves the walls to enter the rooms at 60°, it must enter them at 80°. The *mean* temperature of the air in the walls is thus $\dfrac{60 + 80}{2} = 70°$, and the internal surface would be rather more than 60°, as we assumed. The windows contain 1440 square feet of surface, and will dissipate (293) 1440 × ·53 × 30° = 22896 units per hour, the greater part of which will have to be supplied by the air in the rooms.

(356.) "*Cockle, or Air-stove.*"—It will be seen by (355) that we require 11400 lbs. of air per hour for ventilation, which air has to be heated from 30° to 80°, and will require therefore 11400 × 50° × ·238 = 135660 units of heat. With a cockle such as A in Fig. 112, we should not expect perhaps more than 6000 units usefully from a pound of coals, and in that case the consumption would be $\dfrac{135660}{6000} = 23$ lbs. of coal per hour, requiring $\dfrac{23}{12} = 2$ square feet of fire-grate. The air passing through the cockle, would be heated to 80°, and would become by Table 12, $\frac{150000 \times 1 \cdot 102}{1 \cdot 061} = 156000$ cubic feet per hour, or 2600 per minute, and the velocity given by the chimney (357) being 5 feet per second, or 300 feet per minute, the combined area of the pipes in the cockle, through which the air has to be drawn, must not be less than $\frac{2600}{300} = 8 \cdot 7$ square feet.

A pipe heated externally as in our case gives out no radiant heat to the air within (268), but only by contact. For such a case it is advisable to use a larger volume of air than usual, in order to keep down its temperature as it leaves the furnace, &c., and there will be no loss of effect if we admit a double quantity, the case being quite different to a steam-boiler, &c., where we have shown (101) that a great loss would arise with an extra large volume of air. Then by Table 34, with a double quantity of air, the temperature as it leaves the fire would be 1157°, say 1200°; as it leaves the apparatus it would probably be reduced to 400°, the mean temperature is therefore about 800°, and allowing that the air passing through is heated to 80°, we have a difference of $800 - 80 = 720°$, the ratio for which by Table 82 is about 2·2, and taking the value of A from Table 76 at ·5, we have $\cdot 5 \times 2 \cdot 2 \times 720 = 792$ units per square foot of surface, and the internal area must be $\frac{135660}{792} = 171$ square feet; but with pipes arranged on one another as in Fig. 112, the full external surface is not effectively exposed, probably not more than $\frac{2}{3}$rds can be reckoned on, and hence we require $\frac{171 \times 3}{2}$ = 256 square feet. The size of the pipes must therefore be such as to expose an area to the fire of 256 square feet, and a cross-sectional area of 8·7 square feet for the passage of the air to be heated. These conditions are nearly fulfilled by 8-inch pipes, the cross-sectional area of 8 inches is ·36 square foot, and with 25 pipes we should have $\cdot 36 \times 25 = 9$ square feet. With pipes 9 feet long, allowing that 18 inches is built into the side-walls, the effective length is reduced to 6 feet, and the circumference of 8 inches being about 2 feet, we have $6 \times 2 \times 25 = 300$ square feet, instead of 256 feet as required.

(357.) "*Ventilation.*"—The proportions of the ventilating apparatus must be fixed with reference to summer requirements, when a fire has to be maintained specially for that purpose, and when the air in the draught-chimney, not being so highly heated as in winter, when it receives the waste heat from the cockle, the draught is more feeble. For summer ventilation the damper P is closed, and a fire is maintained night and day in the furnace R at the base of the chimney; this has a closed ash-pit, so that the air feeding the fire is drawn from the body of foul air entering the base of the chimney. The air in the chimney becomes heated, say 50° higher than the atmosphere, and the chimney being 60 feet high, we have by Table 19 a column of hot air say at 82° and 60 feet high, which is counterbalanced by a column at 82° $\frac{60 \times 1 \cdot 00}{1 \cdot 102} = 54 \cdot 4$ feet high, giving $60 - 54 \cdot 4 = 5 \cdot 6$ feet to produce motion, and which by the rule for falling bodies $\sqrt{H} \times 8 = V$, is in our case $\sqrt{5 \cdot 6} \times 8 = 18 \cdot 92$ feet per second. This is the theoretical velocity, which is greatly reduced by the friction of the air in long passages, and by changes of velocity which in practice are unavoidable. These losses, however, cannot be calculated but by many experiments on such chimneys. Péclet finds that where the circuit is long, the real velocity is only ·25 or ·3 of the theoretical, and even where the circuit is short, not more than ·4 (362) (365).

Taking the velocity at ·3 in our case we have $18 \cdot 92 \times \cdot 3 = 5 \cdot 6$, say 5 feet per second, and as we have to pass $\frac{150000}{3600} = 41 \cdot 7$ cubic feet per second, the chimney and main air-passages must have an area of $\frac{41 \cdot 7}{5} = 8 \cdot 34$ square feet, or $144 \times 8 \cdot 34 = 1200$ square inches, and the chimney may be, say 3 feet square.

The area of the channels G will each be 600 square inches; $H = \frac{1200}{18} = 67$ square inches; $N = \frac{1200}{18 \times 3} = 22$; $L = \frac{1200}{6} = 200$ square inches, &c., &c.

(358.) The cost of maintaining the ventilation in winter is nothing, the waste heat from the heating cockle being used; in summer, as we have seen, it is 23 lbs. of coal per hour, or 5 cwt. per day. The ventilation in summer can be adjusted by regulating the fire in R; in winter it may be necessary to regulate the draught by dampers S S, in the main channels G G.

CHAPTER XVI.

EXAMPLES OF BUILDINGS HEATED AND VENTILATED.

As illustrations of the application of the rules given in Chap. XV. for calculating the loss of heat by buildings, and to show how far they agree with observed facts, we may take the following examples which are given by Péclet.

(359.) "*Prison Mazas.*"—This great prison is situated in Paris, and contains cells for the solitary confinement of 1200 prisoners. These cells are arranged in six long buildings, which radiate from a centre, and occupy nearly two-thirds of a circle. The walls are of stone, about 36 feet high, averaging 24 inches thick, and exposing an exterior area of 140000 square feet; the area of the windows is 23400 square feet. For the seven months of the year, during which the building was heated, the mean temperature of the exterior air was 44°, and the interior air 58°, being a difference of 14°, and from these particulars we can calculate the weight of coals required by the building, &c. The ventilation was maintained constantly by a chimney with a fire at its base, as in Fig. 112; this chimney was cylindrical, 7 feet internal diameter and 95 feet high, the mean quantity of air passing through the building was 1059600 cubic feet per hour, and this air had to be heated 14°, from the external to the internal temperatures, and required a further quantity of fuel.

(360.) The conditions of the building are similar to those in (290) and Fig. 84, with one face exposed, and by Table 79 we find that a wall of stone 24 inches thick loses ·284 unit per square foot for 1° difference of internal and external temperature of the air. In our case, therefore, the walls will lose ·284 × 14 × 140000 = 556640 units per hour. The windows were for the most part only 2 feet high, and by the rule in (293) lose ·56 unit per square foot for 1°. The loss in our case is ·56 × 14 × 23400 = 183456 units per hour.

By Table 19 the weight of air at say 42° is ·079 lb. per cubic foot; we have therefore 1059600 × ·079 = 83708 lbs. of air to be heated 14°, and the specific heat of air being ·238, this is equal to 83708 × ·238 × 14 = 278910 units per hour. Collecting these three results we have a total loss of 556640 + 183456 + 278910

= 1019006 units per hour. But part of this heat will be supplied by the animal heat of the prisoners, as explained in (314), where we find that each man will yield 191 units per hour, and we have 191 × 1200 = 229200 units per hour from this source, leaving 1019006 − 229200 = 789806 units to be supplied by the heating-apparatus. Allowing for intermittent firing as in (41), we may take the economic value of coals from Table 31 at 6000 units per pound, and shall require $\frac{789806}{6000} \times 24 = 3160$ lbs. of coal per day. The experimental quantity was found to be 3564: the difference may arise from losses in the vaults, &c., not included in the calculation.*

(361.) During the cold weather of winter it was found that 5280 lbs. of coal were consumed per 24 hours, to maintain the internal air at 59°·3 while the external air was 39°, the difference being 20°·3. We found that for 14° difference the loss was 1019006 units; the loss with 20·3 will therefore be $\frac{1019006 \times 20 \cdot 3}{14} = 1477560$, and deducting the heat emitted by the prisoners, we have 1477560 − 229200 = 124836 units to be supplied by the fuel, requiring $\frac{124836}{6000} \times 24 = 4994$ lbs. of coal, instead of 5280 as per experiment.

(362.) "*Ventilation.*"—For maintaining the ventilation it was found that the mean consumption of coals by the fire at the foot of the chimney was 770 lbs. per day in winter, and 880 lbs. per day for the rest of the year; but for the ventilation, equal to 1059600 cubic feet per hour, the consumption was 44 lbs. per hour in winter, and 55 lbs. in summer.

The temperature of the air in the chimney was not observed, but we can calculate it from the consumption of fuel. By (28) the *total* heat in a pound of coals is 13044 units, nearly the whole of this will be given out to the air passing up the chimney; allowing that 5 per cent. falls from the grate unconsumed as in (90), and that 5 per cent. more is lost by radiation, &c., we have from 44 lbs. of coals 13044 × ·9 × 44 = 516544 units of heat per hour, and as this is carried off by 83708 lbs. of air, and the specific heat of air being $\cdot 238$, we have $\frac{516544}{\cdot 238 \times 83708} = 26^\circ$ increase of temperature, and the air entering at 58° (the temperature of the building) becomes $58^\circ + 26^\circ = 84^\circ$ in the chimney, or $84 - 44 = 40^\circ$ warmer than the exterior air. A 95-feet column of air at 44° is equal by (14) to $95 \times \frac{458 \cdot 4 + 44}{458 \cdot 4 + 84} =$ 88-feet column of air at 84°, and we have $95 - 88 =$ 7-feet head to generate velocity, which will give $\sqrt{7} \times 8 = 21 \cdot 1$ feet per second. The chimney was 7 feet diameter, having an area of 38·5 square feet; but the centre of it was occupied by the iron chimney from the steam-boilers, which was 2 feet 8 inches diameter, having an area of 5·5 square feet. The acting area of the air-shaft was thus reduced to $38 \cdot 5 - 5 \cdot 5 = 33$ square feet, and when discharging 1059600 cubic feet per hour, the velocity of the air entering the chimney must be $\frac{1059600}{33 \times 60 \times 60} = 8 \cdot 92$ feet per second. The real velocity is therefore only $\frac{8 \cdot 92}{21 \cdot 1} = \cdot 423$ of the theoretical velocity, this loss arising from friction, change of velocity by frequent contractions and enlargements in the air-channels, &c., &c., which are practically unavoidable in long circuits; see (161). The mechanical work done by the chimney being proportional to the square of the velocity, is thus reduced to $\cdot 423^2 = \cdot 179$.

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* Péclet brings out the calculated quantity much nearer the experimental quantity than in the above, but he assumes that the temperature of the internal surface of the walls is the same as that of the internal air. This is not a fact, the wall receiving heat from the air, must of necessity be much colder than that air in order to do so. We have shown by Fig. 84 that the wall is about 60 − 54·36 = 5°·64 colder than the internal air.

(363.) "*Mechanical and Chimney Ventilation compared.*"—We may compare mechanical ventilation with that produced by heat, and ascertain the relative economy of the two systems, by calculating the power of an engine and fan capable of doing the same work and the consumption of fuel in the two cases.

By (362) we have seen that with the standard quantity of air the consumption of coals was 44 lbs. per hour during the seven cold months of the year, and 55 lbs. per hour for five months. We have therefore $(44 \times 24 \times 30 \times 7) + (55 \times 24 \times 30 \times 5) = 419760$ lbs. of coals per year, with a heated chimney.

By a high-pressure engine the work in winter costs *nothing*, because the steam after working the engine is used for heating the building.

The mechanical work done is 83708 lbs. of air per hour, or 1228 lbs. per minute, at a velocity of 8·92 feet per second, the head due to which, by the law of falling bodies, is $\left(\frac{8\cdot 92}{8}\right)^2 =$ $1\cdot 243$ feet; we have therefore $\frac{1228 \times 1\cdot 243}{33000} = \cdot 0463$ horse-power. But we have seen that by friction, &c., &c., in the passages the work is reduced to ·179 of the power employed; and as a low-pressure fan and gearing will probably yield only one-third of the force expended on it, we should require $\frac{\cdot 0463 \times 3}{\cdot 179} = \cdot 776$ horse-power. If we allow 10 lbs. of coal per horse-power, this is equal to 7·76 lbs. of coal per hour for seven months of the year only, and the yearly consumption would be $7\cdot 76 \times 24 \times 30 \times 7 = 39110$ lbs. of coal, instead of 419760 lbs. as required by a draught-chimney; the ratio is 1 to 9·3.

This is a favourable case for mechanical ventilation, being a very large one. By the ordinary allowance of 350 cubic feet per head, the air dealt with would suffice for $\frac{1059600}{350} = 3000$ persons, and even for this large number we require an engine of only three-fourths of a horse-power. For small, and even for ordinary cases, the engine would be excessively small, the friction proportionally much greater, and the relative economy of the system less. There is also the practical objection that the engine must work day and night, requiring an extra man for the night-work, &c., and a duplicate engine in case of repairs, &c.

(364.) "*Prison of Provins.*"—This prison was arranged in the same manner as the prison Mazas, but was very much smaller, consisting of one range of buildings for 39 prisoners. The walls were of stone, 24 inches thick, and exposed 11340 square feet of surface. The windows had an area of 1157 square feet, and the air used for ventilation was 120090 cubic feet per hour.

Experiments were made on the consumption of fuel from the 15th of March to the 6th of April, the mean temperature of the *day* was 43° by observation, that of the night was not observed, but is reported to have been very cold, probably it was 8° colder than the day, or 35°; the *mean* temperature of the exterior air would therefore be 39°, and the internal air being maintained at 59°, the difference would be 20°. The consumption of fuel under these circumstances was found to be 807 lbs. of peat per day.

The loss by the walls 24 inches thick will be ·284 units for

1° by Table 79, and we have $\cdot 284 \times 20 \times 11340 = 64411$ units per hour lost by the walls. The windows by (360) will lose $\cdot 56 \times 20 \times 1157 = 12958$ units, and the air $120090 \times \cdot 079 \times \cdot 238 \times 20 = 45348$ units; collecting these three losses, we have $64411 + 12958 + 45348 = 122717$ units per hour. The prisoners will yield $191 \times 39 = 7449$ units, leaving 115268 to be supplied by the fuel.

By (28) and (30) we find the *total* heat in coals and peat to be 13044 and 7151 units respectively, and the economic value of coals being as we have assumed 6000 units, that of peat will be $\frac{6000 \times 7151}{13044} = 3290$ units per lb., and thus we shall require $\frac{115268}{3290} \times 24 = 840$ lbs. of peat per day, whereas experiment gave 807 lbs.

(365.) "*Ventilation.*"—The draught or air chimney was 59 feet high, and 2 feet diameter at the top, its area being 3·14 square feet. An experiment was made when all the fires had been extinguished six hours, the temperature of the air in the chimney would be about the same as that in the building, which was 16° higher than the external air, which being taken at 39° that in the chimney would be 55°. The observed discharge of air under these circumstances was 40052 cubic feet per hour, and the velocity of discharge must therefore be $\frac{40052}{3600 \times 3 \cdot 14} = 3 \cdot 54$ feet per second. For calculating the theoretical velocity we have a column of air 59 feet high at 39°, which is equal in weight to a column $59 \times \frac{458 \cdot 4 + 39}{458 \cdot 4 + 55} = 57 \cdot 11$ feet high at 55°, and we have $59 - 57 \cdot 11 = 1 \cdot 89$ foot head to generate velocity, which will give $\sqrt{1 \cdot 89} \times 8 = 11$ feet per second, whereas the real velocity was only 3·54 feet per second. The ratio in this case is $\frac{3 \cdot 54}{11} = \cdot 322$ to 1, and the mechanical work of the chimney, only $\cdot 322^2 = \cdot 104$ of its theoretical value: see (362).

(366.) "*Church of St. Roch.*"—This church was about 377 feet long, 92 feet wide, and from 50 to 60 feet high. The walls were of stone, 20 inches thick, exposing a surface of 37674 square feet; the windows were about 13 feet high, and exposed an area of 9257 square feet. When the interior air was maintained at 29° above the exterior air, the consumption of coals was 88 lbs. per hour.

The conditions of this building are similar to Fig. 80, the walls being exposed on all sides to cooling influences. The loss of heat for this case is given by Table 78, which for stone walls 20 inches thick may be taken at ·2 for 1°; in our case the loss will be ·2 × 29 × 37674 = 218510 units per hour. By (294) the windows lose ·4 unit for 1°, or in our case ·4 × 29 × 9257 = 107385 units, and the loss from both sources is 325895 units per hour. There will also be another loss of heat by a large but unknown volume of air, which enters through the heating apparatus by openings in the floor, and passes out by innumerable crevices in the leaden casements, the glass being very loosely fitted; it will therefore escape at the temperature of the glass, and if we know that temperature we can calculate the volume of air which passes through the building.

(367.) "*Temperature of the Walls and Glass in Windows*"—To give distinctness to the question, we will assume that the internal air is at 59°, and the external air at 30°, or 29° difference. We can now calculate the temperatures of the two surfaces of the walls by the rules in (282) and (283). Taking the value of R from Table 71 at ·736, of A from Table 74 at ·398, and of C from Table 77 at 13·7, Q will be 1 134 and the rule

$$\frac{Q\,(E\,A\,T + C\,T') + A\,C\,T}{C\,(2\,A + R) + (E\,A\,Q)} = t \text{ becomes in our case}$$

$$\frac{1\cdot134\,(20 \times \cdot398 \times 59 + 13\cdot7 \times 30) + \cdot398 \times 13\cdot7 \times 59}{13\cdot7 \times (2 \times \cdot398) + (\cdot736) + (20 \times \cdot398 \times 1\cdot134)}$$

= 44° which is the temperature of the interior surface of the wall. The temperature of the exterior surface may now be found by the rule $\frac{C\,t + Q\,E\,T'}{C + Q\,E} = t'$, which in our case becomes

$$\frac{(13\ 7 \times 44) + (1\cdot134 \times 20 \times 30)}{13\cdot7 + (1\cdot134 \times 20)} = 35^\circ\cdot27.$$

The temperature of the glass in the windows will be given by the rule in (294) $t'' = \dfrac{\dfrac{(T - t) \times A}{A + R} + t + T'}{2}$ which in our case becomes $\dfrac{\dfrac{(59 - 44) \times \cdot43}{\cdot43 + \cdot5948} + 44 + 30}{2} = 40^\circ\cdot15$, and this is also the temperature of the air at exit.

(368.) "*Volume of Air.*"—We can now calculate the volume of air passing through the building, for as it departs at 40°·15, it must have been cooled by the walls from 59° to 40·15 = 18°·85, and the weight of air which in doing that would yield to the walls and windows the required quantity of 325895 units, must be $\frac{325895}{18\cdot85 \times \cdot238} = 70342$ lbs., or $\frac{70342}{\cdot0791} = 88930$ cubic feet at 42°. This air departing at 40°·15, or 10°·15 above the external air, will carry off $70342 \times 10\cdot15 \times \cdot238 = 170000$ units of heat, which added to the amount lost by the walls and windows gives a total of $325895 + 170000 = 495895$ units per hour, requiring $\frac{495895}{6000} = 82\cdot6$ lbs. of coal, instead of 88 lbs. as per experiment.

(369.) But we might have calculated approximately both the volume of air and its temperature at exit, and therefore of the glass in the windows from the known quantity of coals consumed. 88 lbs. of coals give $88 \times 6000 = 52800$ units of heat, and as the *whole* of this heat is given out to *the air* as it passes the hot-water pipes and enters the building, and that air being heated 29°, its weight must be $\frac{528000}{29 \times \cdot238} = 76500$ lbs. This air has to yield 325895 units to the walls, &c., and to do that must be cooled down $\frac{325895}{76500 \times \cdot238} = 17^\circ\cdot9$, it therefore departs from the building at $59^\circ - 17^\circ\cdot9 = 41^\circ\cdot1$, and this is also the temperature of the glass.

The volume of air and temperature of the glass as thus calculated, differs but little from those found by the former method, which no doubt is the more correct of the two.

(370.) Taking the volume of air at 88930 cubic feet per hour, and allowing as in (312) 215 cubic feet per head, this would suffice for $\frac{88930}{215} = 4100$ people; the ordinary numbers present are 2000 to 4000; on fête-days, 4000 to 6000; and on grand occasions, 6000 to 8000. The capacity of the church is about 113000 cubic feet of air; it is therefore renewed about 8 times per hour.

(371.) "*Time required to heat the Building to the Standard Temperature.*"—It was found that to heat the building to the standard temperature of 29° above the external air, required eight days of continuous firing day and night, and it will be instructive to see how far this agrees with calculation.

The walls contain 63576 cubic feet of stone, weighing by Table 46 about 156 lbs. per foot, or $63576 \times 156 = 9917856$ lbs., and the specific heat by Table 1 being about ·21585, they will require 9917856 × ·21585 = 2140770 units of heat to raise their temperature 1°. We found in (367) that when the standard temperature was attained, the two surfaces are 44° and 35°·27 respectively, the *mean* temperature of the wall is therefore $\frac{44 + 35 \cdot 27}{2}$ = 39°·625, or 9°·625 above the external temperature, and this is the amount of heat which the walls must receive before the standard temperature can be attained. We found that for 1° they required 2140770 units, they will therefore require 2140770 × 9·625 = 20626320 units to bring them up to the standard.

(372). The time in which this quantity can be supplied is in our case governed by the maximum power of the heating-apparatus. This was proved with an external temperature of 21°, for in that case the internal temperature could not be maintained higher than 50°, being a difference of 29°, or the same as at the standard internal temperature of 59°, with external air at 30°. The apparatus could therefore yield only 218510 units per hour *to the walls* as in (366), and to do that, the temperature of the walls must be 15° colder than the air as we have seen in (367), where the respective temperatures were 59° and 44°. This difference would be constant throughout, so that the walls being 30° at the commencement, the internal air would be heated by the apparatus to 45°, when further increase of temperature would be arrested by the walls absorbing the heat, and as their temperature increased, so would that of the air be progressively and simultaneously increased, and the difference of 15° would be maintained throughout.

(373.) But while heat was thus received by the internal surface, a considerable amount would be dissipated by the external surface. At the commencement, when the walls had the same temperature as the external air, the loss would be 0, progressively increasing to 218510 units per hour when the walls were heated to the standard temperature; for when that is attained, the external surface dissipates the same amount as the internal surface receives, and the temperature remains stationary. The *mean* loss is therefore $\frac{218510 + 0}{2}$ = 109255 units per hour, so that while 218510 units were received by the interior surface, 109255 units would be dissipated by the external surface, and 109255 units would be stored up in the walls; and to obtain 20626320 units, the time required to bring them to the standard temperature would be $\frac{20626320}{109255 \times 24} = 7 \cdot 87$ days, practically the same as by experiment, which, as we have stated, was eight days.

(374.) With a powerful heating-apparatus this time might be shortened; but if the maximum temperature of the internal air be limited to 59° (the standard temperature), which is almost a practical necessity, as a higher temperature would be intolerable, the *time* will be governed by the absorbing power of the walls, irrespective of the power of the apparatus beyond a certain point. With the apparatus we have considered, the air could not be heated more than 15° above the temperature of the walls; but with one of greater power the difference might be 29° or $59° - 30°$, so that the air could be raised at once to the standard temperature of 59°, and the walls would then absorb $\frac{218510 \times 29}{15} = 422424$ units; but as the temperature of the walls increased, that of the air being fixed at 59°, the difference would become less and less, until when the standard temperature was attained, it would be reduced to 15°, and the amount of heat received to 218510 units per hour, the mean is therefore $\frac{422424 + 218510}{2} = 320467$ units per hour. The outer surface would still lose at a mean rate of 109255 units, and the amount stored in the walls being thus $320467 - 109255 = 211212$ units per hour, we should require $\frac{20626320}{211212 \times 24} = 4 \cdot 07$ days to attain the standard temperature.

(375.) If the building is not to be occupied during the time its walls are being raised to the standard temperature, the maximum temperature of the internal air need not be limited to 59°, and in that case the time might be still further shortened. For instance, with air at 88°, the difference at the commencement would be $88 - 30 = 58°$, or double that in the last case, which was 29°, and the walls would absorb at first $422424 \times 2 = 844848$ units per hour; the *mean* heat received would then be $\frac{844848 + 218510}{2} = 531679$, and the mean amount dissipated being as before 109255, the amount stored in the walls would be $531679 - 109255 = 422424$ units, and the time required to bring the building up to the standard temperature

$\frac{20626320}{422424 \times 24} = 2 \cdot 03$ days. In this case the fire would have to be put out some time before the hour of worship, to allow the air to cool down to 59°, and the time for this may be approximately calculated.

(376.) The church contained 113000 cubic feet of air, or $113000 \times \cdot 0733 = 82829$ lbs. of air, which in cooling down 29° from 88° to 59°, would yield $82829 \times 29 \times \cdot 238 = 57168$ units. The hot-water apparatus contained 15886 lbs. of water at 248°, and in cooling that down 189°, or to 59°, we shall have $15886 \times 189 = 3002454$ units; so that to cool down to the standard temperature we must absorb $57168 + 3002454 = 3574140$ units. During the cooling process the *mean* temperature of the internal air is $\frac{88 + 59}{2} = 73° \cdot 5$, being $73 \cdot 5 - 30 = 43° \cdot 5$ colder than the external air. With 29° difference, we found (366) the loss by the windows to be 107385 units, the temperature of the glass being 40°·15, or 10°·15 colder than the external air; but with 43°·5 difference, as in our case, the temperature of the glass by the rule in (294) is 43°·2, or 13°·2 colder than the external air, and the loss by the windows and the air in the two cases will be proportional to those differences. In the former case these losses were $107385 + 170000 = 277385$; in our present case it will therefore be $\frac{277385 \times 13 \cdot 2}{10 \cdot 15} = 360730$ units per hour. The walls at this time are absorbing their standard quantity of 218510 units; so that altogether the cooling influences are equal to $360730 + 218510 = 579240$ units per hour, and the time in which the internal air, &c., will be cooled down is $\frac{3574140}{579240} = 6 \cdot 17$ hours. If the heating apparatus had consisted of *steam*-pipes instead of hot-water pipes, we should not have had the body of water to cool down, and the air alone would have been cooled to the standard temperature in $\frac{571686}{579240} =$ 1 hour nearly.

(377.) It should be observed that the time required to bring the walls up to the standard temperature is simply proportional to the power of the heating-apparatus, but that the mean heat consumed, and consequently the consumption of fuel, follows a much lower ratio. The maximum power of the apparatus in the three cases we have considered was 218510, 422424, and 844848 units per hour respectively, the ratio being 1 . 2 . 4

nearly, but the *mean* heat given out in the three cases was 218510, 320467, and 531679, the ratio being 1, 1·46, and 2·43.

(378.) We have assumed in the foregoing that the area of the internal surface absorbing heat was equal to that of the external surface dissipating it; and this is nearly true in our case, the internal area being increased by massive stone pillars, buttresses, &c., supporting the roof.

(379.) "*Time required to Cool the Building.*"—When once heated, a considerable time would be required to cool the whole building down to the external temperature. If the air passing through it be stopped by closing the inlet-openings, heat would be dissipated only by the walls and windows, and we should lose 325895 units at first, which would be progressively reduced to nothing at the end of the time. The mean loss would therefore be $\frac{325895 + 0}{2} = 162947$ units per hour, and the heat of the building would be dissipated in $\frac{20626320}{162947 \times 24} = 5 \cdot 27$ days.

If the circulation of air be permitted as usual, the mean rate of loss would be $\frac{528000 \times 0}{2} = 264000$ units per hour, and the building would cool down to the external temperature in $\frac{20626320}{264000 \times 24} = 3 \cdot 25$ days.

(380.) "*Heating-Apparatus.*"—The heating-apparatus consisted of a boiler about 12-horse power, with hot-water pipes 5½ and 4¾ inches diameter. They were laid in channels with brick sides, under the floor of the church in the usual way. The total area of the hot-water pipes was 1774 square feet, they therefore gave out $\frac{495875}{1774} = 279 \cdot 5$ units per square foot: (see (368.) The temperature of the water as it left the boiler was 248°, and as it returned 216°, its mean temperature was therefore 232°, or $232° - 29° = 203°$ above the external air. By Table 66, a 6-inch *enclosed* pipe yields 286 units per square foot, for 200° difference. The case of a pipe enclosed in a channel is quite different to that of a pipe freely exposed.

(381.) When a pipe is enclosed, the walls receive radiant heat from it, and their temperature is raised until they give out to the air in contact the same amount as they are receiving from the pipe, when further increase of temperature is arrested, and it remains stationary. With brick walls and a steam-pipe at 232° the temperature of the walls would be 158°. At that temperature, being $232 - 158 = 74°$ colder than the pipe, the ratio (299) of heat lost by radiation is, by Table 80, $1\cdot9$, and the value of R from Table 71 being for bricks $\cdot736$, the walls will receive $\cdot736 \times 74 \times 1\cdot9 = 103$ units per square foot per hour. Then by (301) a wall, say 2 feet high, will have $\cdot528$ for the value of A by Table 74, and the difference of temperature of the air and of the wall being $158 - 30 = 128°$, the ratio by Table 81 is $1\cdot486$, and the loss by contact of air is $\cdot528 \times 128 \times 1\cdot486 = 100\cdot3$ units per hour, nearly the same amount as was received by radiation, and showing that the temperature of 158° is very nearly correct.

(382.) A steam-pipe 5 inches diameter, heated to 232°, exposed to radiant walls at 158°, and to air at 30° (see 302), will lose $(\overset{\text{R.}}{\cdot7} \times \overset{\text{Diff.}}{74°} \times \overset{\text{Ratio.}}{1\cdot9}) + (\overset{\text{A.}}{\cdot544} \times \overset{\text{Diff.}}{202°} \times \overset{\text{Ratio.}}{1\cdot65}) = 279\cdot7$ units per square foot per hour, which by an accidental coincidence is almost exactly the amount found by experience, which we have seen to be $279\cdot5$ units.

If this same pipe had been freely exposed to air and radiant objects both at 30°, the difference of temperature would have been $232° - 30° = 202°$ in both cases, and the loss $(\overset{\text{R.}}{\cdot7} \times \overset{\text{Diff.}}{202} \times \overset{\text{Ratio.}}{1\cdot51}) + (\overset{\text{A.}}{\cdot544} \times \overset{\text{Diff.}}{202} \times \overset{\text{Ratio,}}{1\cdot65}) = 394\cdot8$ units, and an enclosed pipe therefore gives only $\frac{279\cdot7}{394\cdot8} = \cdot708$, or say 70 per cent. of the amount lost by a pipe freely exposed (303).

CHAPTER XVII.

WIND, FORCE OF, AND EFFECT ON VENTILATION, ETC.

(383.) "*Influence of the Wind in Ventilation, &c.*"—The effect of the wind on the draught-power of chimneys to furnaces, &c., is well-known by experience. Let A B C, Fig. 13, be a tube freely exposed to the action of the wind, A B being vertical; if now the wind moves in the direction C to B, it will enter at C and pass out at A; but if the wind moves from B to C, it will enter at A and pass out at C.

It will be seen, that this is analogous to the case of a boiler-house on a plane, with a chimney at one end, as in Fig. 15, the whole being freely exposed to the wind on all sides, and air admitted by openings at the end C; but here the air in the chimney is highly heated, and a powerful upward draught thereby created, still it will be retarded when the wind sets from B to C, and assisted and accelerated when in the contrary direction. Table 54 shows the draught-power of chimneys in calm air, in inches of water pressure, by which we can compare the power of the draught, with that of the wind.

(384.) According to the experiments of Hutton, the force of wind at moderate velocities varies as the 2·04 power of the velocity: he also found that a plane 32 square inches in area, and moving at a velocity of 12 feet per second, experiences a resistance from the wind of ·841 ounce. From these data we obtain the formulæ.

$V^{2\cdot04} \times \cdot001487 = P$ and $\frac{P}{\cdot001487} = V^{2\cdot04}$, in which V = the velocity in feet per second, and P = the pressure in lbs. per square foot; Table 89 has been calculated by these rules.

(385.) Taking as an illustration a chimney 80 feet high, we find by Table 54 that the draught-power is equal to ·585 inch of water, and by Table 89 this is equal to a wind having a velocity of about 42 feet per second, which is a very brisk breeze, and if the boiler-house is freely exposed to its force, the draught of the chimney would be destroyed when the wind was blowing with that strength in the direction B C, and *doubled* when in the direction C B. In such a case the doors or other openings at the end C should be closed, and others opened at the end B or or at the side D.

(386.) To obviate the adverse action of the wind, and to utilize its power of increasing the draught, we may use a movable cowl, whose action and principle may be illustrated by Fig. 14, in which we have a tube like Fig. 13 surmounted by a cowl, movable round the centre of the tube A B by the action of the wind itself, which is assisted by the vane E. When the wind is in the adverse direction B C, the cowl A D opposes and balances the effect of the wind on the lower branch, and if both are equally exposed to its action, the one would simply neutralize the other, and there would be no internal current in either direction, but if, as is usually the case in practice, B C is sheltered by adjacent buildings, &c., and A D is fully exposed, the cowl will always be the controlling power, creating in all cases an upward current, irrespective of the action of heat on the internal air, and the adverse action of the wind is not only annulled, but reversed and made to augment the draught.

TABLE 89.—Of the FORCE and VELOCITY of WIND, according to Hutton's Experiments.

Pressure.		Velocity.		Character of the Wind according to Rouse and Lind.
In lbs. per square Foot.	In Inches of Water.	In Feet per Second.	In Miles per Hour.	
·1	·01926	7·87	5·37	Gentle Wind.
·25	·0481	12·90	8·79	Pleasant ditto.
·5	·0963	16·18	11·03	Fresh Breeze.
1	·193	24·33	16·60	Brisk Gale.
2	·385	34·17	23·30	Very brisk ditto.
3	·578	41·69	27·77	
4	·770	48·00	32·73	
5	·963	53·55	36·51	High Wind.
6	1·155	58·55	39·92	
7	1·348	63·15	43·06	
8	1·541	67·43	45·97	
9	1·733	71·43	48·70	Very high ditto.
10	1·926	75·22	51·28	
11	2·118	78·82	53·74	
12	2·311	82·20	56·07	
13	2·504	85·54	58·32	
14	2·696	88·70	60·48	Storm or Tempest.
15	2·889	91·76	62·56	
16	3·081	94·70	64·57	
17	3·274	97·56	66·52	
18	3·467	100·30	68·41	
19	3·659	103·00	70·25	
20	3·852	105·70	72·03	Great Storm.
25	4·815	117·80	80·36	
30	5·778	128·9	87·97	Hurricane.
35	6·741	139·0	94·77	
40	7·704	148·7	101·4	Very great ditto.
45	8·667	157·2	107·2	
50	9·630	165·6	112·9	Most violent ditto.
55	10·593	177·5	118·3	
60	11·556	181·0	123·4	

(387.) The effect of the vane E may be greatly augmented by making it with double blades, as in Fig. 99; the experiments of Hutton show that the force of wind acting obliquely on a plane, is given by the rule ($\text{Sine}^{1{\cdot}842\ \text{Cosine}}$), and he gives a table calculated by that rule, from which we obtain with a single blade, the forces ·003, 006, &c., &c., as per Table 90, the force at a right angle or 90° being 1·0. It has been found by the numbers in this Table, that the best angle for the blades is 38° with the axis, or 76° with each other, as in Fig. 100. To show the superior efficacy of a double-bladed vane, say that the wind was at an angle of 1° with the axis, then with a vane having blades at the best angle instead of impinging on both blades at 38°, we should evidently have it at 37° on one, and 39° on the other, and by Hutton's Table the forces with these angles would be ·507 and ·555, and the force tending to turn the vane ·555 − ·507 = ·048 instead of ·003, as with a single blade of the same area, or $\frac{\cdot 048}{\cdot 003}$ = 16 times greater.

TABLE 90.—Of the RELATIVE POWER of WIND-VANES, with Single and Double Blades.

	Angle of Wind with the Axis.							
	1°	2°	3°	4°	5°	10°	20°	30°
	Ratio of the Turning Force.							
Single Blade ..	·003	·006	·010	·014	·018	·046	·156	·347
Double Blades, angle 38° with axis	·048	·095	·141	·185	·228	·431	·739	·910
Ratio of effect, single blade being 1	16	15·8	14·1	13·2	12·7	9·4	4·7	2·6

(388.) Table 90 shows that the superiority of the double blade over the single is greatest at small angles, where in fact it is most wanted. Such a vane would not only be more sensitive but also more steady, or less subject to oscillation. Angles greater or less than 38° are not so effective, the turning power at 15° would be about ½, and at 5° about ⅕th of that at 38°.

(389.) "*Maximum Force of the Wind.*"—In this country, the force of the wind seldom exceeds 50 lbs. per square foot even in our great storms; it has been known, however, occasionally to exceed that amount. In the great hurricane of January, 1868, the anemometer at Birmingham, which was graduated up to 60 lbs., had its pencil driven far beyond that pressure, the maximum force was estimated at 70 or 80 lbs. per square foot.

(390.) We will calculate the force of the wind capable of overturning the chimney 80 feet high, shown by Fig. 44. There are two forces which resist the wind, namely, the weight of the whole chimney and the cohesion of the mortar. The chimney contains 1747 cubic feet of brickwork, which weighs by Table 46 115 lbs. per foot, giving a total weight of 200000 lbs. The cohesion of good old mortar (14 years old) is 60 lbs. per square inch, and the area of the section of base being $81^2 - 36^2 = 5265$, the resistance of the cohesion of the mortar will be $5265 \times 60 = 315900$ lbs., which added to the weight of the chimney makes a total of $200000 + 315900 = 515900$ lbs. If the materials were incapable of crushing, the chimney would turn on that edge of its base remote from the wind, but in truth that point would be somewhere between the centre and the edge, and the chimney would resist fracture, partly by the crushing strain and partly by the cohesion assisted by the weight. By analogy with other materials broken transversely, we know that the result is very nearly the same as if the neutral axis coincided with the edge, and the force of cohesion only came into play. Admitting this, the force of 515900 lbs. acts with a leverage equal to half the diameter of the base, or 40·5 inches. The centre of effort of the wind is at the centre of gravity of the surface exposed to it; the easiest way of finding the centre of gravity in our case, is by *cutting out* an outline of the chimney in drawing paper, &c., of equable thickness, and balancing it on the point of a needle. We thus find the centre of effort in our case to be 36 feet, or 432 inches above the base, and the surface area of one side of the chimney being 440 square feet, the force of the wind that would overturn it would be $\frac{515900 \times 40 \cdot 5}{432 \times 440} =$ 110 lbs. per square foot, which as we have seen is greater than any wind in this country. With very bad mortar, the force would be much less, for instance, without mortar altogether, the force of the wind would be $\frac{200000 \times 40 \cdot 5}{432 \times 440} = 42 \cdot 6$ lbs. per square foot.

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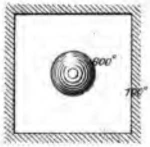
Fig. 1.



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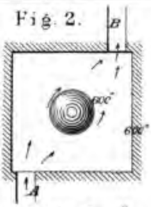

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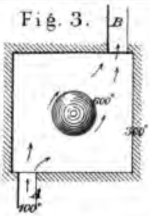

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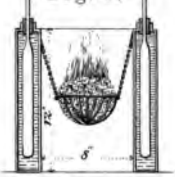

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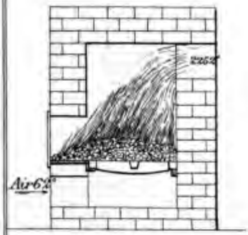

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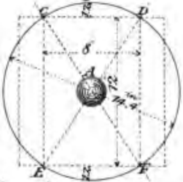

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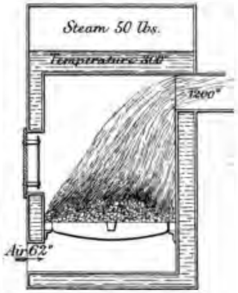
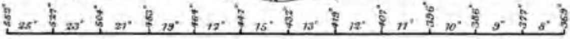


Fig. 9.



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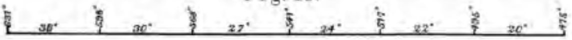

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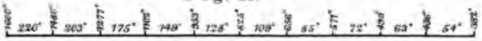

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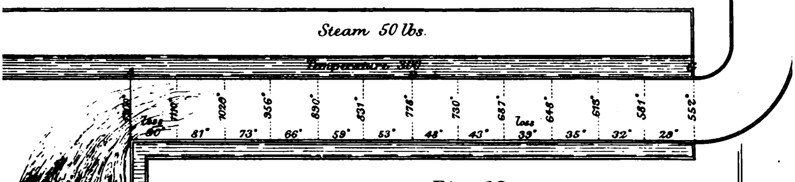
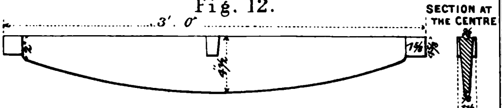

Steam 50 lbs.
Temperature 300°
1200°
1110°
Loss 90°
1020° 81° 956° 73° 890° 66° 831° 59° 778° 53° 730° 48° 687° 43° 648° Loss 39° 615° 35° 581° 32° 552° 29°

Fig. 12.



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E & F N. Spon, 48, Charing Cross, London.

For 35 and 50 H.P. See Plate 3.

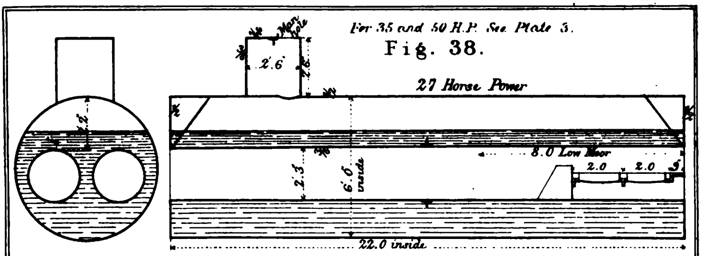

Fig. 38.

27 Horse Power

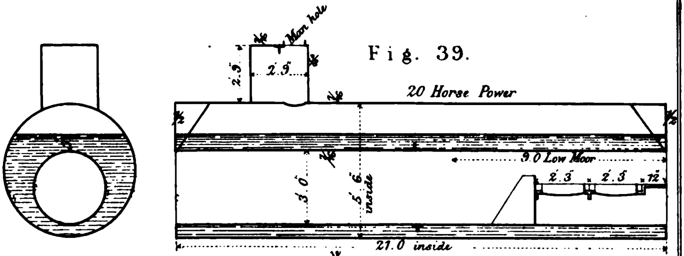

Fig. 39.

20 Horse Power

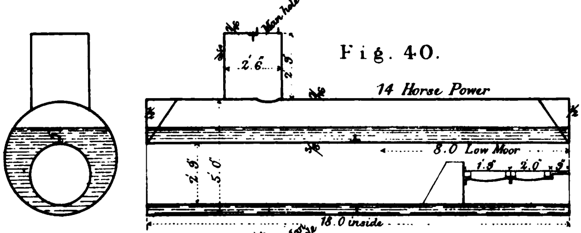

Fig. 40.

14 Horse Power

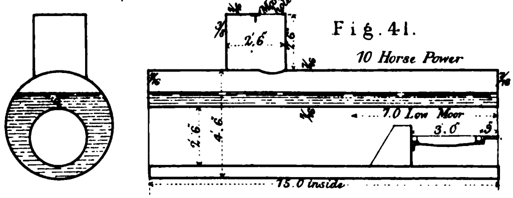

Fig. 41.

10 Horse Power

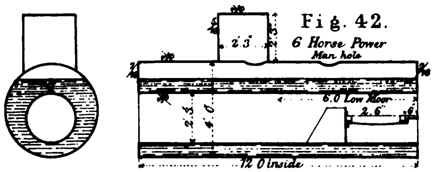

Fig. 42.

6 Horse Power

For 4 H.P. See Plate 5.

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Fig. 43.

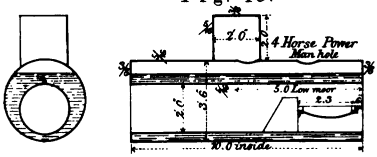
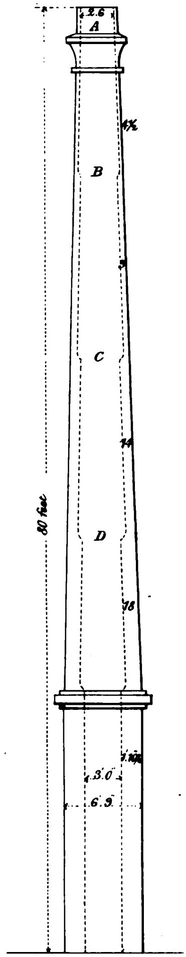


Fig. 44.



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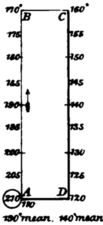

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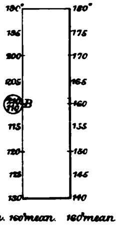

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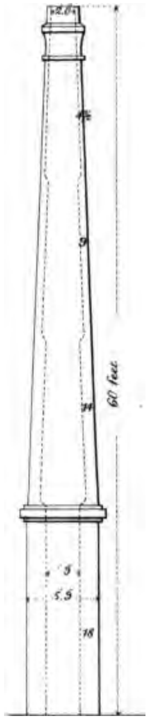

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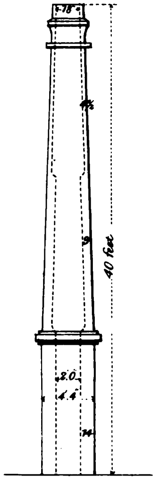

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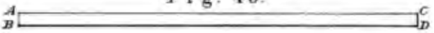
Fig. 49.



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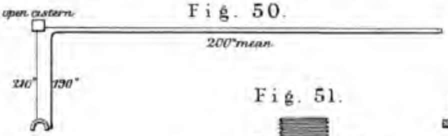

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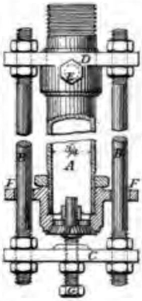

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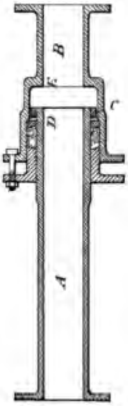

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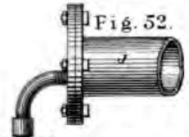
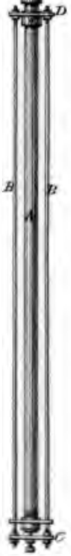

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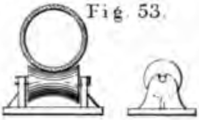

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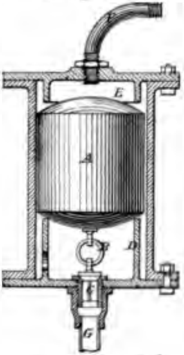

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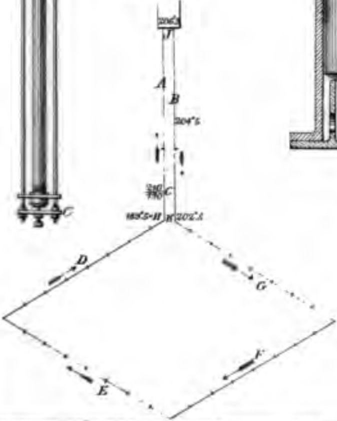

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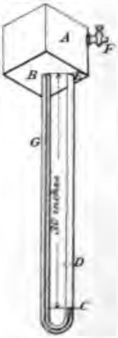

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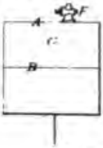

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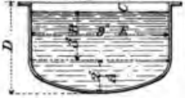
Fig. 59.



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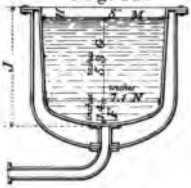

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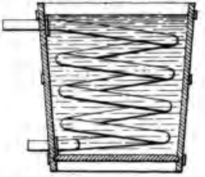

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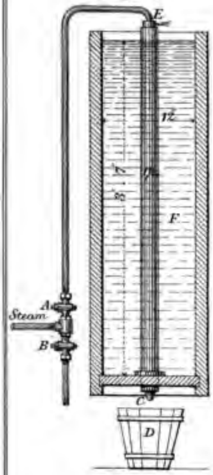

Fig. 63 SECTION.

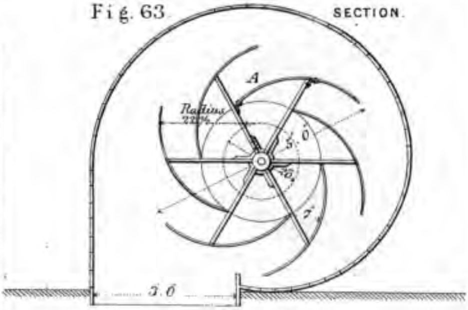

Fig. 64.
SECTIONAL PLAN.

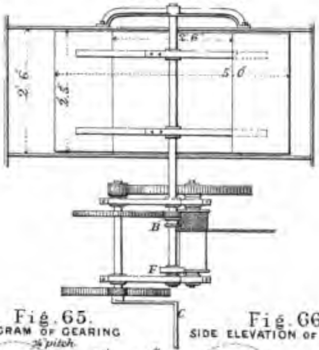

Fig. 65.
DIAGRAM OF GEARING

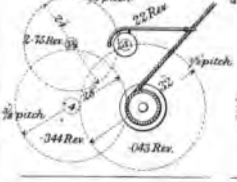

Fig. 66.
SIDE ELEVATION OF FRAMING

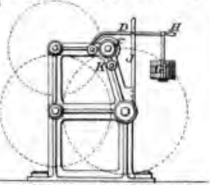

Fig. 67

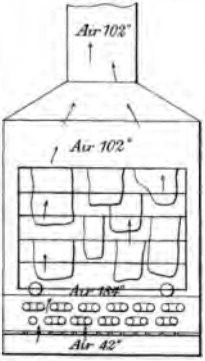

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Fig. 68.
LONGITUDINAL SECTION.

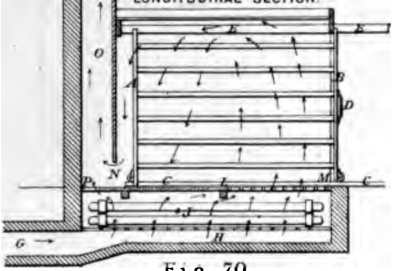


Fig. 69.
CROSS SECTION.

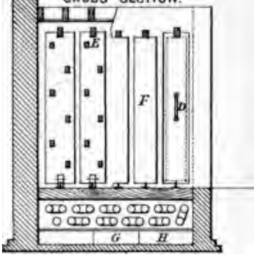
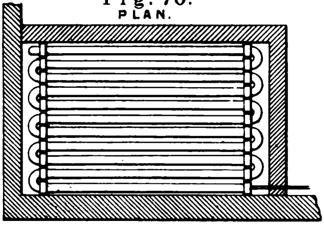


Fig. 70.
PLAN.



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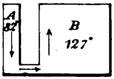

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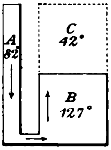

Fig. 73.

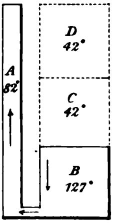

Fig. 74.
Loss by Radiation

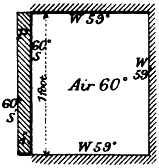

Fig. 75.
Loss by Cold Air.

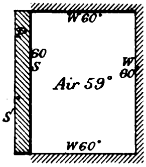

Fig. 76.
Loss by Conduction

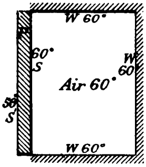

Fig. 77.
Loss by R×A×C

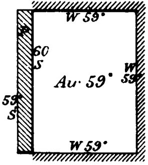

Fig. 78.

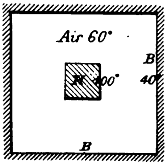

Fig. 79.

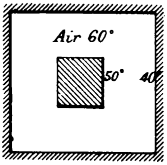

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Fig. 80.

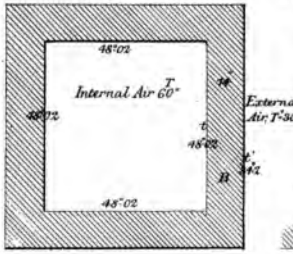
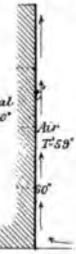


Fig. 81.



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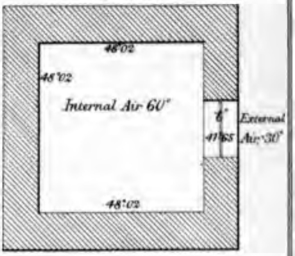

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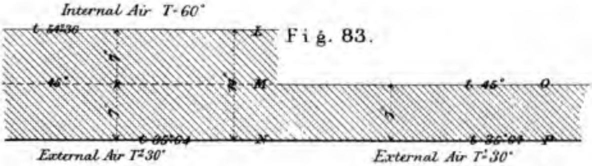

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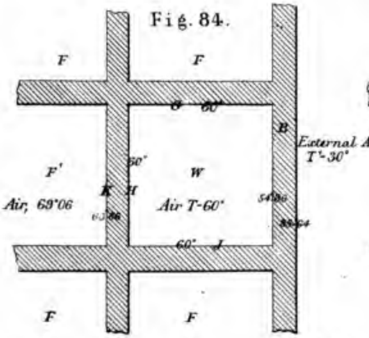

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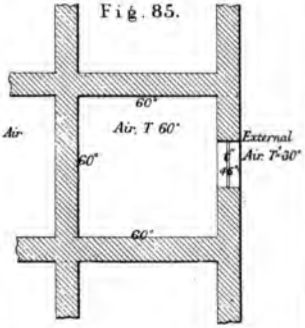

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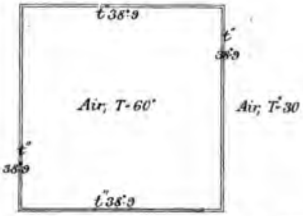

[illegible] & [illegible], lith. 37, Castle St Holborn.

E. & F. N. Spon, 48, Charing Cross, London.

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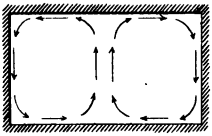

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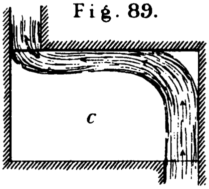

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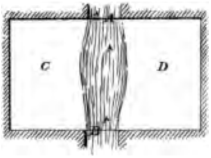

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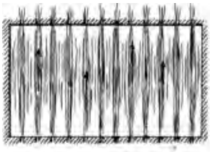

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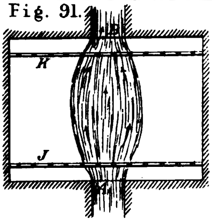

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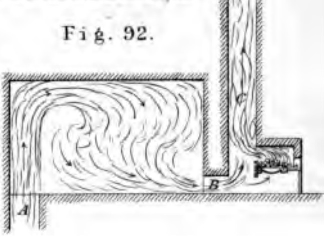

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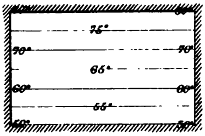

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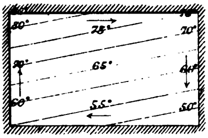

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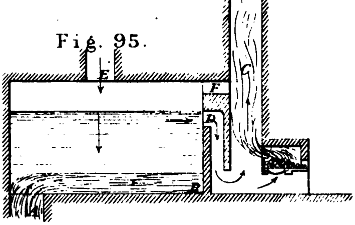
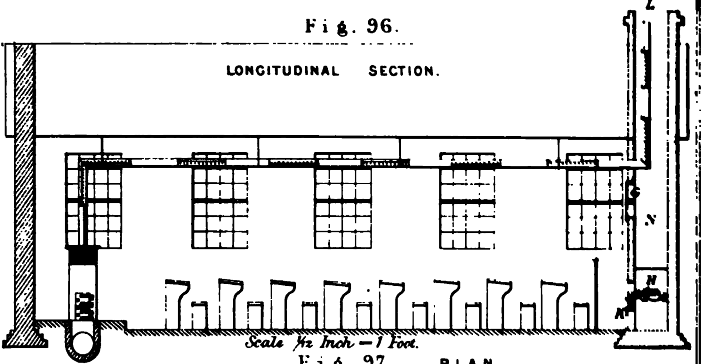


Newman & Alexander, lith. 4, Castle St. Holborn.

E & F N. Spon. 48, Charing Cross, London.

Fig. 96.

LONGITUDINAL SECTION.



Scale 1/12 Inch = 1 Foot.

Fig. 97. PLAN.

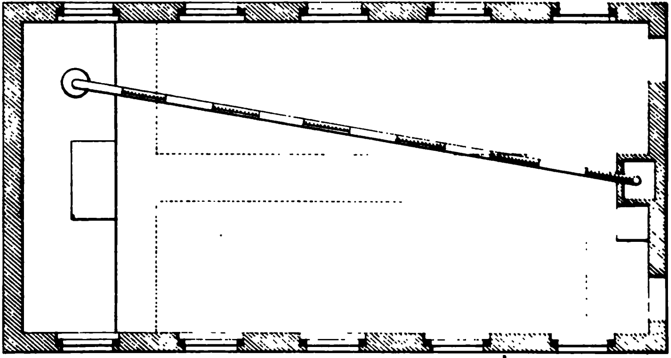
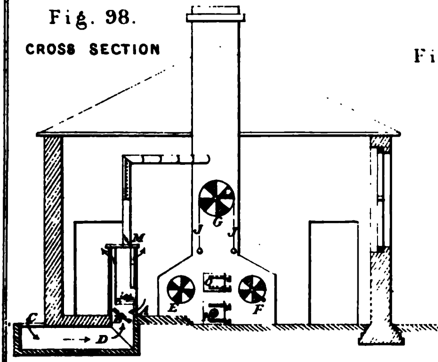


Fig. 98.

CROSS SECTION



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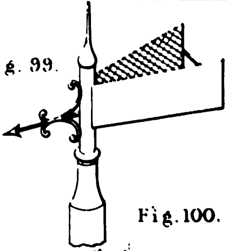

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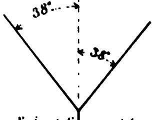

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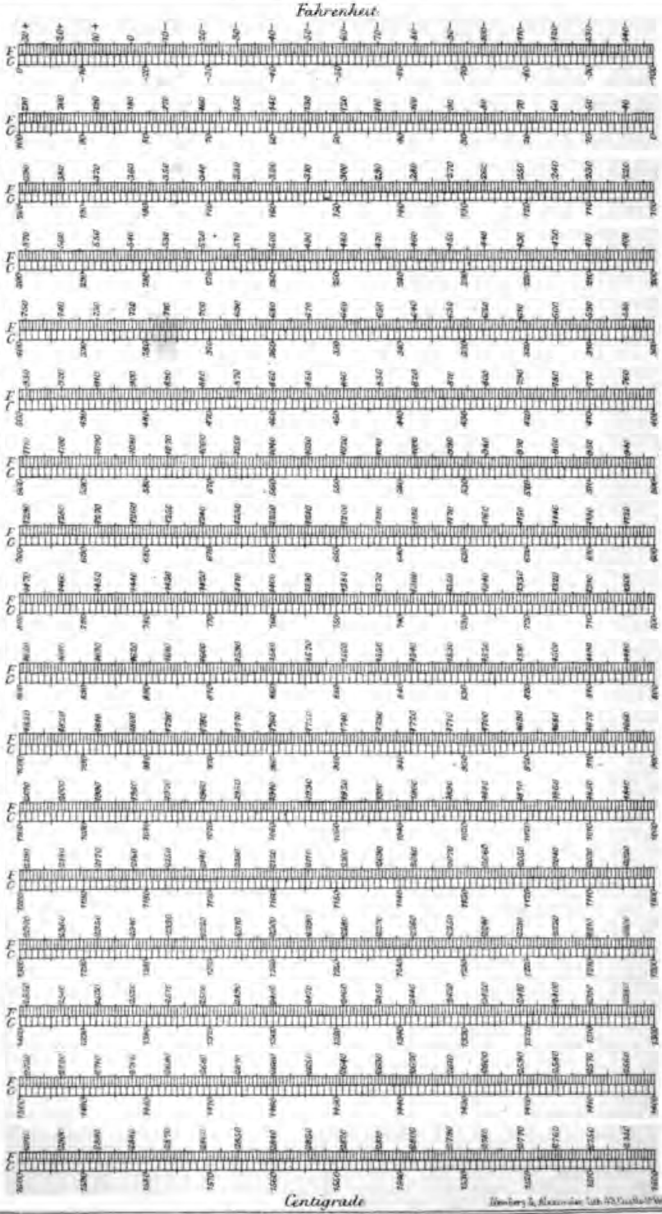
E & F. N Spon. 48, Charing Cross London.

TABLE OF THE CORRESPONDENCE OF THE FAHRENHEIT AND CENTIGRADE THERMOMETERS.



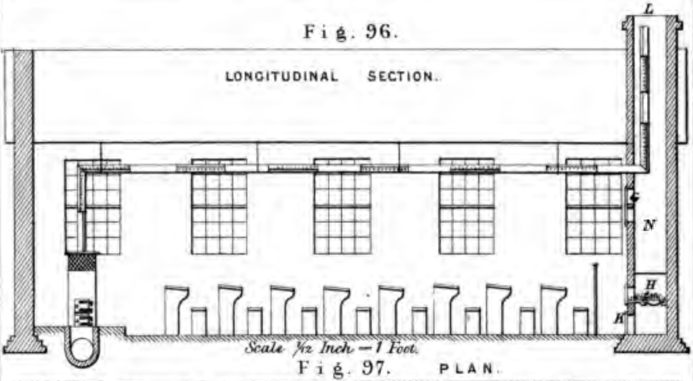
Hanbury & Alexander Lith 43 Castle St Holborn

E. & F. N. Spon, 48, Charing Cross, London.

PLATE 11.

Fig. 96.

LONGITUDINAL SECTION.



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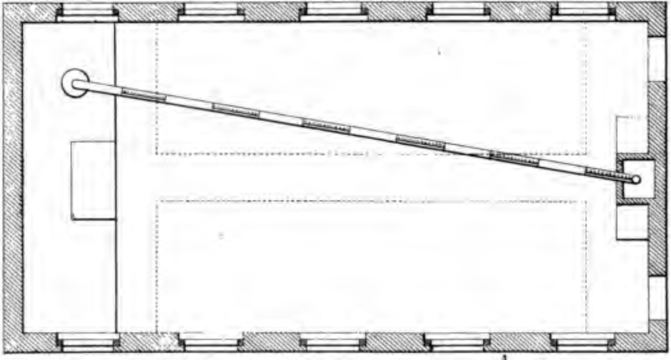

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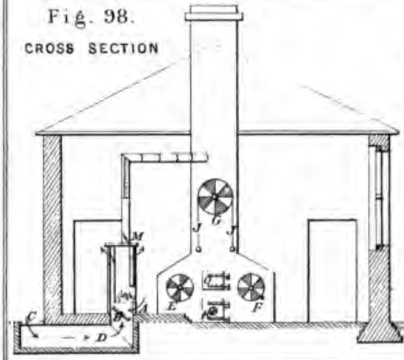
CROSS SECTION



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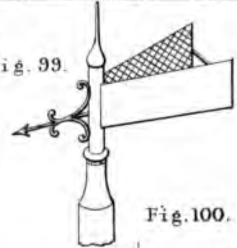

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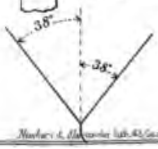


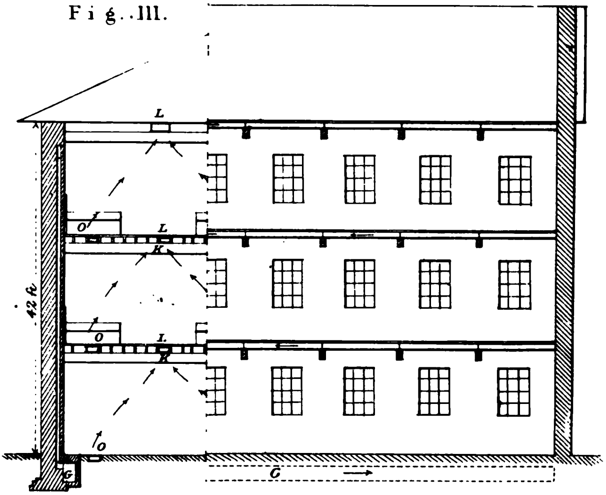
Newman & Alexander lith. 43, [illegible] Holborn.

E & F. N Spon, 48, Charing Cross, London.

PLATE 14.

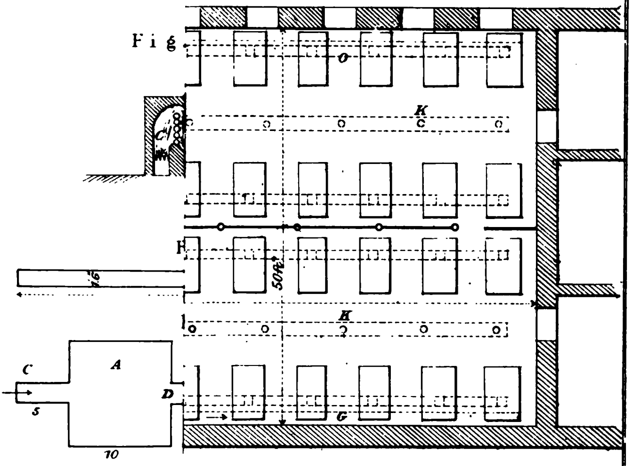
ECTION.

Fig..III.



D.

Fig

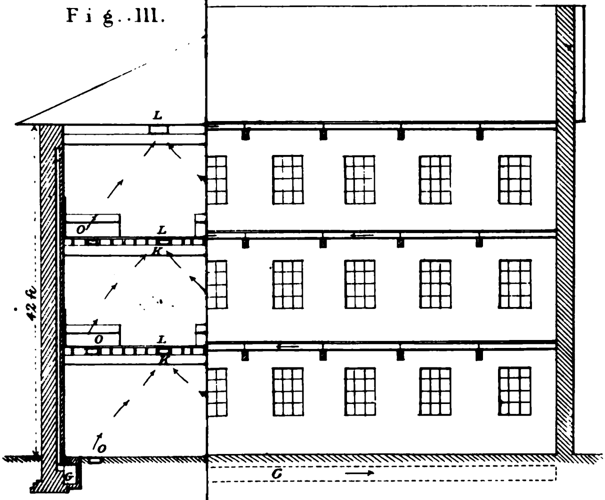


Newbery & Alexander, lith 43 Castle St Holborn

PLATE 14.

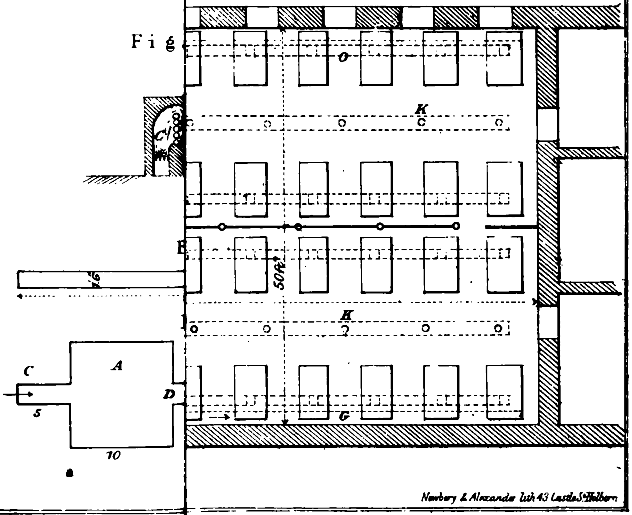
ECTION.

Fig..III.



D.

Fig



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